Renewable and Sustainable Energy Reviews 13 (2009) 1151–1184

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

Effect of engine parameters and type of gaseous fuel on the performance of dual-fuel gas diesel engines—A critical review

B.B. Sahoo [a], N. Sahoo [b], U.K. Saha [b,*]

[a] Centre for Energy, Indian Institute of Technology, Guwahati 781039, India
[b] Department of Mechanical Engineering, Indian Institute of Technology, Guwahati 781039, India

ARTICLE INFO

Article history:
Received 7 January 2008
Accepted 20 August 2008

Keywords:
Combustion
Carbon dioxide
Dual-fuel
Efficiency
Emissions
Gaseous fuel

ABSTRACT

Petroleum resources are finite and, therefore, search for their alternative non-petroleum fuels for internal combustion engines is continuing all over the world. Moreover gases emitted by petroleum fuel driven vehicles have an adverse effect on the environment and human health. There is universal acceptance of the need to reduce such emissions. Towards this, scientists have proposed various solutions for diesel engines, one of which is the use of gaseous fuels as a supplement for liquid diesel fuel. These engines, which use conventional diesel fuel and gaseous fuel, are referred to as 'dual-fuel engines'. Natural gas and bio-derived gas appear more attractive alternative fuels for dual-fuel engines in view of their friendly environmental nature. In the gas-fumigated dual-fuel engine, the primary fuel is mixed outside the cylinder before it is inducted into the cylinder. A pilot quantity of liquid fuel is injected towards the end of the compression stroke to initiate combustion. When considering a gaseous fuel for use in existing diesel engines, a number of issues which include, the effects of engine operating and design parameters, and type of gaseous fuel, on the performance of the dual-fuel engines, are important. This paper reviews the research on above issues carried out by various scientists in different diesel engines. This paper touches upon performance, combustion and emission characteristics of dual-fuel engines which use natural gas, biogas, producer gas, methane, liquefied petroleum gas, propane, etc. as gaseous fuel. It reveals that 'dual-fuel concept' is a promising technique for controlling both NO_x and soot emissions even on existing diesel engine. But, HC, CO emissions and 'bsfc' are higher for part load gas diesel engine operations. Thermal efficiency of dual-fuel engines improve either with increased engine speed, or with advanced injection timings, or with increased amount of pilot fuel. The ignition characteristics of the gaseous fuels need more research for a long-term use in a dual-fuel engine. It is found that, the selection of engine operating and design parameters play a vital role in minimizing the performance divergences between an existing diesel engine and a 'gas diesel engine'.

Contents

* Corresponding author. Tel.: +91 361 2582663; fax: +91 361 2690762/2582699.
E-mail address: saha@iitg.ernet.in (U.K. Saha).

Abbreviations: ATDC, after top dead centre; BMEP, brake mean effective pressure; bsfc, brake specific fuel consumption; BTDC, before top dead centre; C, centigrade; CA, crank angle; DI, direct injection; DG, diesel genset; FD, fossil-diesel; HRR, heat release rate; IDI, indirect injection; J, joule; kW, kilo Watt; m, mass, meter; NO_x, oxides of nitrogen; P, pressure; Pa, pascal; rpm, revolutions per minute; T, torque; UBHC, unburned hydrocarbon; BSU, Bosch smoke unit; BTE, brake thermal efficiency; CR, compression ratio; BDC, bottom dead centre; CO, carbon monoxide; db, decibel; deg., degree; h, hour; HC, hydrocarbon; IT, injection timing; K, Kelvin; l, litre; M, mega; N, Newton; NG, natural gas; ppm, parts per million; rev, revolutions; RHR, rate of heat release; SO_x, oxides of sulphur; TDC, top dead centre.

1364-0321/$ – see front matter
doi:10.1016/j.rser.2008.08.003

1. Introduction

National interest in generating alternative fuels for internal combustion (IC) engines continues to be strong due to environmental concerns and/or the uncertainties associated with the future availability of fossil fuel. Mostly, the interest lies in identifying alternative sources of fuel energy supply. The motivation for the need for diversification has prompted the research worldwide into potential alternative sources of fuel energy for IC engines. One feature of alternative sources of fuel energy is that they are well suited for decentralised development to meet the needs for social and economic progress, especially in rural communities where fossil fuels may be difficult or expensive to obtain. Natural gas, bio-derived gas and liquids appear more attractive in view of their friendly environmental nature [1,2].

The gaseous fuels are getting more positive response from researchers and end-users compared to past because of current unfolding developments. Gas is clearly the fossil fuel of least environmental impact. When burnt, it produces virtually insignificant SO_x and relatively little NO_x, the main constituents of acid rain, and substantially less CO_2, a key culprit in the greenhouse debate, than most oil products and coal [3].

Most combustion devices are easily adaptable to the use of gaseous fuels for power production. High compression ratio piston engines are generally suitable when high thermal efficiency is desired. Gaseous fuels have high octane numbers, and therefore, suitable for engines with relatively high compression ratio. Gaseous fuels also promise to be suitable for higher compression engines, since it is known that they resist knock more than conventional liquid fuels, as well as producing less polluting exhaust gases if appropriate conditions are satisfied for its mixing and combustion. Therefore, it is more economical and of environmental advantage to use gaseous fuel in diesel engines that use the 'dual-fuel concept' [4,5].

Due to limited resources of fossil fuels, alternative solutions have been proposed by many scientists. The "dual-fuel concept" is one of them that use both conventional diesel and gaseous fuels. The lower running costs and the use of alternative fuel sources with dual-fuel engine operation have attracted many investigators to use this engine in different areas of interest. The earliest experiments on dual-fuel system were performed by Cave in 1929, and Helmore and Sokes in 1930, in which burning hydrogen is induced as a secondary fuel in diesel engines. When hydrogen burn completely, there was reduction of liquid fuel load and saving of 20% diesel fuel [6]. However, at that time, the dual-fuel engine was not used commercially due to its mechanical complexity and rough running caused by auto-ignition and knocking at relatively low compression ratios. In 1939, the first commercial dual-fuel engine, fuelled by town gas or other types of gaseous fuels, was produced by the National Gas and Oil Engine Co. in Great Britain. This type of engine was relatively simple in operation and was mainly employed in some areas where cheap stationary power production was required. During the Second World War, scientists in Great Britain, Germany and Italy paid more attention to the possible application of dual-fuel engines in civil and military areas due to the shortage of liquid fuels. Some vehicles with diesel engines were successfully converted to dual-fuel running. Different kinds of gaseous fuels, such as coal gas, sewage gas or methane, were employed in conventional diesel engines [7]. After the Second World War, due to economical and environmental reasons, dual-fuel engines have been further developed and employed in a very wide range of applications from stationary power production to road and marine transport, such as in long and short haul trucks and buses. Some conversions from the original compression ignition diesel engines to dual-fuel operation are made by manufacturers utilizing a double plunger system or two pumps in the injection system of the engine to handle the small quantity of diesel fuel required for ignition [8].

Diesel engines, with appropriate relatively simple conversion, can be made to operate on gaseous fuels efficiently. Such engines, which are called 'dual-fuel engines' or 'gas diesel engines', usually have the gaseous fuel mixed with the air in the engine cylinders, either through direct mixing in the intake manifold with air or through injection directly into the cylinder. The resulting mixture after compression is then ignited through the injection of a small amount of diesel fuel (the pilot) in the usual way. This pilot liquid fuel auto-ignites to provide ignition sources for subsequent flame propagation within the surrounding gaseous fuel mixture. Unlike

Nomenclature

Greek letters

η	efficiency
θ	crank angle
$^{\circ}$	degree

Subscripts

d	dual-fuel
g	gas

the spark ignited gas engine, which requires an adequate and uninterrupted gas supply, most current dual-fuel engines are made to operate interchangeably, either on gaseous fuels with diesel pilot ignition or wholly on liquid fuel injection as a diesel engine. Accordingly, a dual-fuel engine tends to retain most of the positive features of diesel operation [9]. Even it surpasses occasionally those of the diesels, producing higher power outputs and efficiencies. This is achieved without significant smoke or particulates emission and with reduced NO_x production [10], while having reduced peak cylinder pressures and quieter operation.

When considering a gaseous fuel for use in existing diesel engines, a number of issues are important. These issues include the engine operating and design parameters, and type of gaseous fuel supply to the engine. The purpose of this review is to discuss the effect of engine operating and design parameters, and type of gaseous fuel on the performance of the gas diesel engine. The engine operating and design parameters include load, speed, compression ratio, pilot fuel injection timing, pilot fuel mass inducted, intake manifold conditions, and type of gaseous fuel. The effects of these parameters on performance, combustion and emission characteristics of dual-fuel diesel engines are also examined. This paper touches upon dual-fuel engines that use natural gas, biogas, producer gas, methane, liquefied petroleum gas, propane, etc. as gaseous fuel. An attempt is made here to review the previous studies to look into further improvement of "gas diesel engines" from the viewpoint of performance, combustion and emission.

2. Dual-fuel concept

Available technologies for reciprocating IC engines are generally divided in two categories: compression-ignition (CI) and spark-ignition (SI) engines. In CI engines (diesel engines), air is compressed at pressures and temperatures at which the injected liquid fuel fires easily and burns progressively after ignition. Whereas, SI engines (Otto engines) that runs according to the Beau de Rochas cycle [11], the carburetted mixture of air and vaporized fuel (high octane index) is compressed under its ignition point and then fired at a chosen instant by an independent means.

In a dual-fuel engine, both types of above combustion coexist together, i.e. a carburetted mixture of air and high octane index gaseous fuel is compressed like in a conventional diesel engine. The compressed mixture of air and gaseous fuel does not auto-ignite due to its high auto-ignition temperature. Hence, it is fired by a small liquid fuel injection which ignites spontaneously at the end of compression phase. The advantage of this type of engine is that, it uses the difference of flammability of two used fuels. Again, in case of lack of gaseous fuel, this engine runs according to the diesel cycle by switching from dual-fuel mode. The disadvantage is the necessity to have liquid diesel fuel available for the dual-fuel engine operation [12].

3. Modification in internal combustion gas engines

A SI gas engine modification is comparatively easy as it is already designed to operate on air–fuel mixture with spark ignition. The basic modification needed in this type of engine is the provision of a gas–air mixer instead of the carburettor. The engine control is performed by the mixture supply variation, i.e. throttle valve position like in the case with petrol fuel.

In dual-fuel gas diesel engines, the normal diesel fuel injection system still supplies a certain amount of diesel fuel. The engine however induces and compresses a mixture of air and gaseous fuel that is prepared in the external mixing device. The compressed mixture is then ignited by energy from the combustion of the diesel fuel spray, which is called pilot fuel. The amount of pilot fuel needed for this ignition is between 10% and 20% of the operation on diesel alone at normal working loads and the amount differs with the point of engine operation and its design parameters. During part load engine operation, the fuel gas supply is reduced by means of a gas control valve. However, a simultaneous reduction of the air supply decreases the air quantity induced. Hence, the compression pressure and the mean effective pressure of the engine are decreased. This would eventually lead to a drop in power and efficiency. The drastic reduction in the compression conditions might even become too weak for the mixture to effect self-ignition. Therefore, dual-fuel engines should not to be throttled/controlled on the air side.

Ideally, there is a need for optimum variation in the liquid pilot fuel quantity used any time in relation to the gaseous fuel supply so as to provide for any specific engine the best performance over the whole load range desired [12]. Usually, the main goal, for both emissions and economic reasons, is to minimize the use of the diesel fuel and maximize its replacement by the cheaper gaseous fuel throughout the whole engine load range. The dual-fuel engine can operate effectively on a wide range of different gaseous fuels while maintaining the capacity for operation as a conventional diesel engine. Normally, the change over from dual-fuel to diesel operation and vice versa can be made automatically, even under load conditions [3].

4. Combustion processes in gas diesel engine and conventional diesel engine

The combustion processes in CI engines running on pure diesel fuel can be divided into four stages as shown in Fig. 1. They are, A–B: period of ignition delay; B–C: premixed (rapid pressure rise) combustion; C–D: controlled (normal) combustion; and D–E: late combustion. Point 'A' is the start of fuel injection and 'B' for start of combustion.

However, the combustion processes in gas-fumigated dual-fuel engines using pilot injection have been identified to take place in five stages as shown in Fig. 2. They are the pilot ignition delay (AB), pilot premixed combustion (BC), primary fuel ignition delay (CD), rapid combustion of primary fuel (DE) and the diffusion combustion stage (EF).

Ignition delay (AB) of injected pilot fuel exists longer than the pure diesel fuel operation. This is due to the reduction in oxygen concentration resulting from gaseous fuel substitution for air. The pressure rise (BC) is moderately low as compared to pure diesel

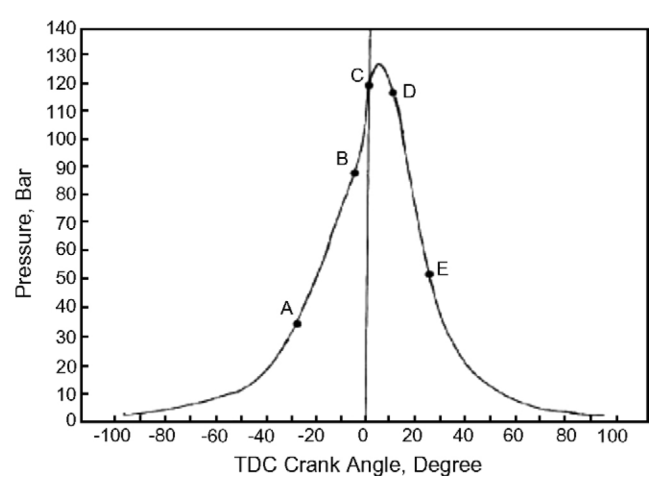

Fig. 1. Details of combustion processes in diesel engine [13].

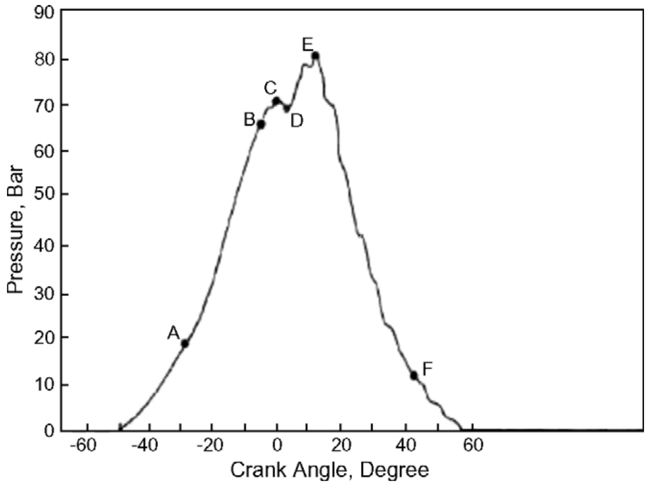

Fig. 2. Dual-fuel pilot injection pressure–crank angle diagram [13].

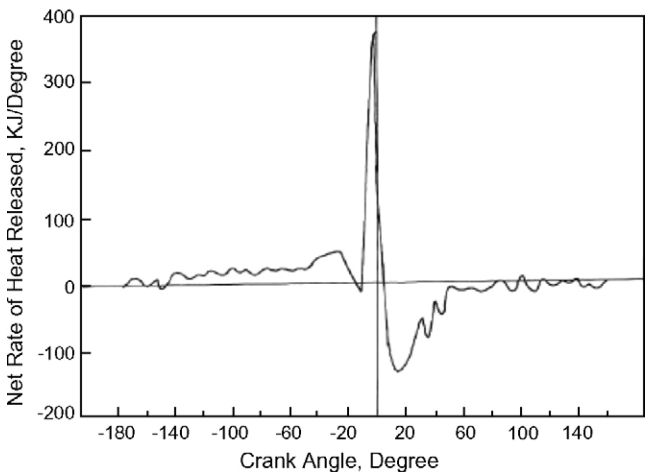

Fig. 3. Heat release diagram of diesel fuel operation (engine speed: 3000 rev/min, engine torque output, N-m: 9.65) [13].

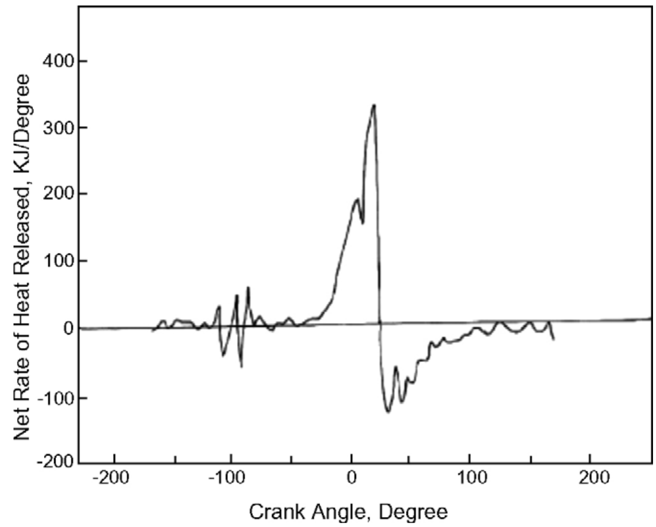

Fig. 4. Dual-fuel pilot injection heat release diagram (engine speed: 3600 rev/min, engine torque output, N-m: 5.15) [13].

fuel operation due to the ignition of small quantity of pilot fuel. Again, there is a finite time lag between the development of the first and second pressure rises due to a longer ignition delay of gas–air mixture, a result of the high self-ignition temperature. However, this ignition delay is short as compared with the initial delay period due to the pilot fuel injection. The pressure decreases slowly (CD) until the actual combustion of the fumigated gas starts. The phase of combustion (DE) is very unstable because it started with flame propagation that has been initiated by the spontaneous ignition of pilot fuel. The pressure rise here does not cause any operating problem since it occurs in an increasing cylinder volume. Diffusion combustion stage (EF) starts at the end of rapid pressure rise and continues well into the expansion stroke. This is due to the slower burning rate of gaseous fuel and the presence of diluents from the pilot fuel. Some gas–air mixture may escape combustion under this phase due to low oxygen concentration, valve overlap, flame quenching on the walls or the effects of crevices. The success of this phase primarily depends on the length of ignition delay.

The heat release diagram of diesel fuel operation (Fig. 3) shows an apparent negative heat release prior to the main start of combustion. This is due to the cooling effect of the injected liquid fuel. This effect is not apparent with the dual-fuel combustion processes as pre-oxidation of the gaseous fuel starts before pilot fuel injection (Fig. 4). The pilot fuel is then flattened the heat release rate within this region. Fig. 4 shows the initial heat release

Table 1
Summary of engine type and dual-fuel used in the experimental investigation by various researcher(s).

Researcher(s)	Test engine	Pilot fuel	Primary fuel
Nwafor [2,13,37]	Petter model AC1 single cylinder, air-cooled, high speed, IDI, four-stroke diesel engine	Diesel	NG
Bari [40]	Two cylinder four stroke cycle diesel engine (16.8 kW at 1500 rpm, Model-2105 Nang Chang Company, China), water cooled, naturally aspirated with double swirl combustion chamber	Diesel	Biogas
Singh et al. [30]	A naturally aspirated multi cylinder DG with matching alternator	FD	Producer gas
Mansour et al. [12]	Naturally aspirated, V-8 Deutz FL8 413F four cycle diesel engine	Diesel	NG
Papagiannakis and Hountalas [19]	Single cylinder, naturally aspirated, four stroke, air cooled, direct injection, high speed, Lister LV1 DI diesel engine with a bowl in piston combustion chamber	Diesel	NG
Selim [14]	Ricardo E6 single cylinder variable compression IDI diesel engine	Diesel	Compressed NG
Selim [4]	Ricardo E6 single cylinder variable compression IDI diesel engine	Diesel	CH_4, CNG, LPG
Nwafor and Rice [38]	Petter model AC1 single cylinder, air-cooled, high speed, IDI, four-stroke diesel engine	Diesel	NG
Henham and Makkar [3]	Two-cylinder, four-stroke, water-cooled, IDI Lister Petter LPWS2 diesel engine	Gasoil	Biogas
Papagiannakis and Hountalas [24]	Single cylinder, naturally aspirated, four stroke, air cooled, DI, high speed, Lister LV1 diesel engine with a bowl in piston combustion chamber	Diesel	NG
Krishnan et al. [35]	Single-cylinder DI, CI engine	Diesel	NG
Kusaka et. al. [39]	Water cooled, 4-stroke-cycle, and 4-cylinder conventional DI diesel engine	Diesel	NG
Uma et al. [16]	Direct injected six cylinder, vertical, four stroke engine with mechanical injector	Diesel	Producer gas
Badr et al. [9]	Two single cylinder, 4-stroke, water cooled, DI, normally aspirated laboratory dual-fuel engines	Diesel	Methane
Abd et al. [33,36]	Single cylinder, four stroke, water cooled engine (Ricardo E6)	Diesel	Methane, propane

due to pilot injection. This figure also indicates that dual-fuel combustion is continued well into the expansion stroke [13].

5. Performance of "gas diesel engines"

Gaseous fuels are considered to be good alternative fuels for passenger cars, truck transportation and stationary engines that can provide both good environmental effect and energy security [4]. However, as some of the engine operating and design parameters namely, load, speed, compression ratio, pilot fuel injection timing, pilot fuel mass, inlet manifold condition, composition of gaseous fuel candidates vary, the performance of the dual-fuel gaseous engines are affected. Numerous studies have been carried out by many researchers to examine the effect of the above-mentioned parameters on the emission, performance and combustion characteristics of dual-fuel engines. These investigations by the researchers are conducted in different test engines with various gaseous primary fuels and pilot fuels. The type of test engine and fuels for dual fuel operation used by several researcher(s) are given in Table 1.

5.1. Effect of engine load

The effect of load on combustion noise for the diesel and dual fuel engine (Fig. 5), at an engine speed of 1200 rpm is examined by Selim [14]. The pressure rise rate (combustion noise) for the diesel engine increases slightly when the load increases. For the dual-fuel mode, the combustion noise also increases when the load increases and is always higher than that for the diesel fuel case. Combustion noise for the diesel case is about 4 bar/°C A and it only fluctuates slightly around this value. For the dual-fuel engine, it increases from 4 bar/°C A at a load of 4.5 N-m to 15.5 bar/°C A at 18.5 N-m. Increasing the load at constant speed results in an increase in the mass of gaseous fuel admitted to the engine, since the pilot mass injected remains constant at all loads. This increase in the mass of methane then causes an increase in the ignition delay period of pilot diesel which then auto-ignites and starts burning the gaseous fuel at a higher rate of pressure rise. This is also shown by Nielsen et al. [15] on dual-fuel engine where natural gas is admitted in the inlet air manifold.

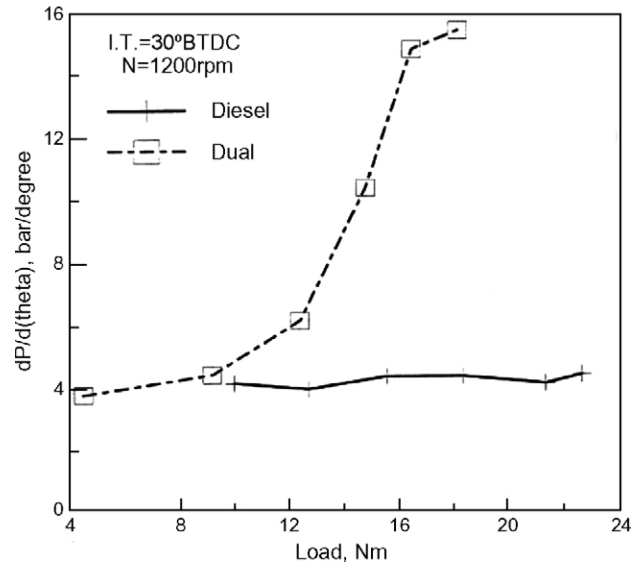

Fig. 5. Effect of engine load on pressure rise rate for the diesel and dual-fuel engines [14].

Concentration of pollutants is investigated with diesel alone and dual-fuel mode (producer gas) at different loads (10, 20, 30 and 40 kW) by Uma et al. [16] and is given in Table 2. NO_x emissions in dual-fuel mode are lower than the emissions from diesel engine in diesel alone mode. SO_2 levels are low in dual fuel mode. This is due to low sulphur content in biomass fuel. The CO emission in dual-fuel mode is higher than that of diesel alone. High concentration of CO in the dual-fuel exhaust is an indication of incomplete

Table 2
Concentration of pollutants from diesel engine in diesel alone and dual-fuel mode [16].

Parameter	Load (kW)							
	10		20		30		40	
	Diesel	Dual-fuel	Diesel	Dual-fuel	Diesel	Dual-fuel	Diesel	Dual-fuel
CO (ppm)	181	635	207	640	284	734	323	904
	174	661	218	681	275	693	303	941
	172	672	232	696	294	705	336	922
CO_2 (ppm)	3.1	6.2	4.2	7.1	5.7	9.2	6.1	11.0
	3.0	6.0	4.1	7.2	5.8	9.1	6.1	10.9
	3.2	6.2	4.2	7.2	5.9	9.3	6.2	11.1
HC (ppm)	109	119	132	141	180	182	270	262
	112	127	125	136	188	178	271	288
	103	113	116	121	187	192	276	284
CH_4 (ppm)	7.0	18	8.4	24	10.2	21	12.1	32
	7.2	12	9.2	27	10.1	32	12.3	30
	7.0	14	8.7	30	10.2	25	11.8	36
SO_2 (ppm)	4.6	1.1	5.4	1.2	6.8	1.5	9.6	2.3
	4.2	1.2	5.0	1.2	6.9	1.6	9.6	2.3
	3.9	1.2	5.8	1.3	8.4	1.9	10.3	1.9
NO_2 (ppm)	172	93	230	140	279	170	412	240
	181	98	229	145	282	161	403	249
	188	101	224	137	279	171	405	253
Particulates (mg/m^3)	22	18	26	24	29	24	33	28
	20	20	20	27	23	18	27	22
	27	22	32	24	32	29	36	40

Table 3
Fuel consumption and SEC of diesel engine in diesel alone and dual-fuel mode [16].

Load (kW)	Fuel consumption			SEC (MJ/kWh)	
	Diesel mode	Dual-fuel mode		Diesel mode	Dual-fuel mode
	Diesel (kg/h)	Diesel (kg/h)	Producer gas (Nm^3/h)		
10	5.3	1.9	57	22.8	34
20	7.2	1.3	66	15.5	18
30	9.8	1.5	81	14.0	15
40	12.2	3.7	112	13.1	16

combustion. At part load condition, concentration of CO increases. This also suggests the need for lower load limit for dual-fuel operation. At part load condition, the specific energy consumption (SEC) also increases. High CO emission in dual-fuel mode operation is due to combination of factors such as low heating value of gas, low adiabatic flame temperatures, and low mean effective pressures. Additionally, the engines are not actually designed for producer gas operation. But these issues can be resolved if efforts are focused on development of new engine designs for producer gas operation. Apart from injector design, other parameters such as compression ratio, ignition advance, combustion chamber design, etc. will have to be optimized to produce low CO emissions. HC emissions in dual-fuel mode are little lower than HC emissions in diesel alone mode.

The SEC increases with decreasing load both in diesel alone and dual-fuel mode (Table 3). This implies the considerable efficiency loss at low-load condition. SEC in dual-fuel mode is higher than the diesel mode throughout the tested load condition. Increased SEC indicates the efficiency reduction in the dual-fuel mode. This is due to reduced heating value of the producer gas–air mixture and drop in the pressure of the gas entering the air inlet and lower flame velocity. The results are similar to the earlier studies [17,18] which also reported de-rating of diesel engine operated in dual-fuel mode.

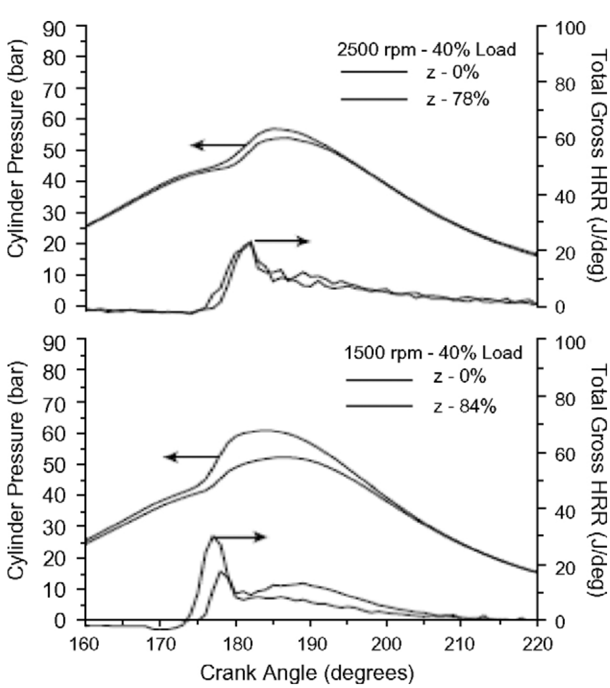

Fig. 6. Cylinder pressure and total heat release traces under normal diesel ($z = 0\%$) and dual-fuel ($z > 0\%$) operation for 1500 and 2500 rpm engine speed at 40% load [19].

An experimental investigation has been conducted by Papagiannakis and Hountalas [19] to examine the effect of dual fuel combustion on the performance and pollutant emissions of a DI diesel engine. The engine has been properly modified to operate under dual-fuel operation. The air inlet temperature is kept 23 °C for all cases. Measurements are taken at four different engine loads corresponding to 20%, 40%, 60%, and 80% of full load and three engine speeds of 1500, 2000 and 2500 rpm under both normal diesel operation (only diesel fuel) and dual-fuel operation with NG and pilot injection of diesel fuel. Under dual-fuel operation, an effort has been made to keep the pilot amount of diesel fuel constant, while the power output of the engine is adjusted through the amount of gaseous fuel. Referring to Fig. 6, the term 'z' refers to the percentage of gaseous fuel. At part engine load, cylinder pressure under dual-fuel operation diverges from the respective values under normal diesel operation. The lower cylinder pressures observed under dual-fuel operation during the compression stroke are the result of the higher specific heat capacity of the NG–air mixture. The total heat release rate under dual-fuel operation is slightly higher compared to the one under normal diesel operation (Fig. 6); revealing late combustion of the gaseous fuel. But, the effect on the cylinder pressure is small since it is in the expansion stroke. At high engine load (Fig. 7), the cylinder pressure traces under dual-fuel operation diverge again from the respective values under normal diesel operation during the compression stroke and the initial stages of combustion. This difference is again more evident at low engine speed where the combustion rate under

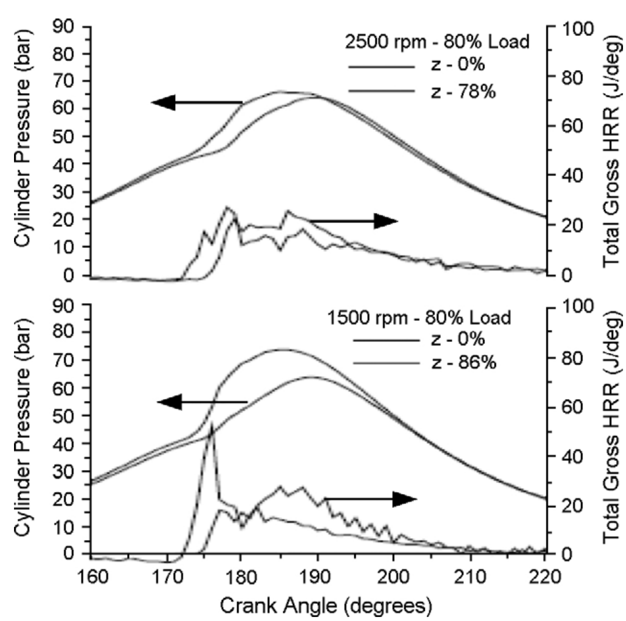

Fig. 7. Cylinder pressure and total heat release traces under normal diesel ($z = 0\%$) and dual-fuel ($z > 0\%$) operation for 1500 and 2500 rpm engine speed at 80% load [19].

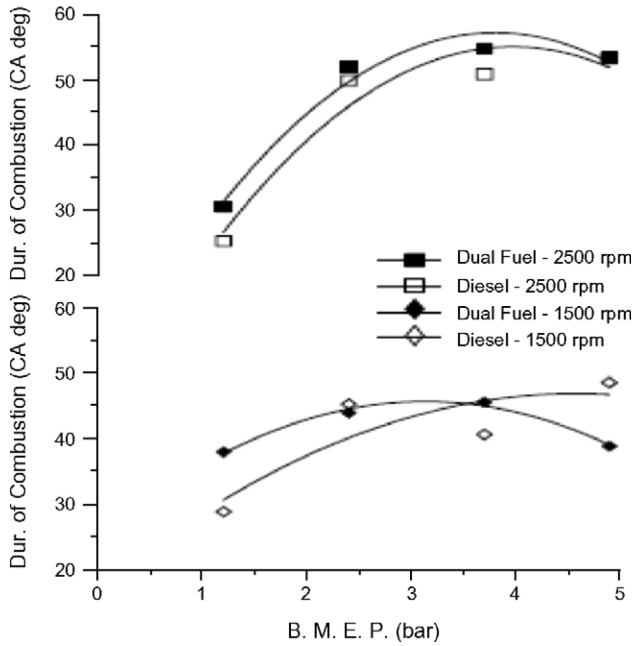

Fig. 8. Variation of combustion duration under normal diesel and dual-fuel operation as function of load at 1500 and 2500 rpm engine speed [19].

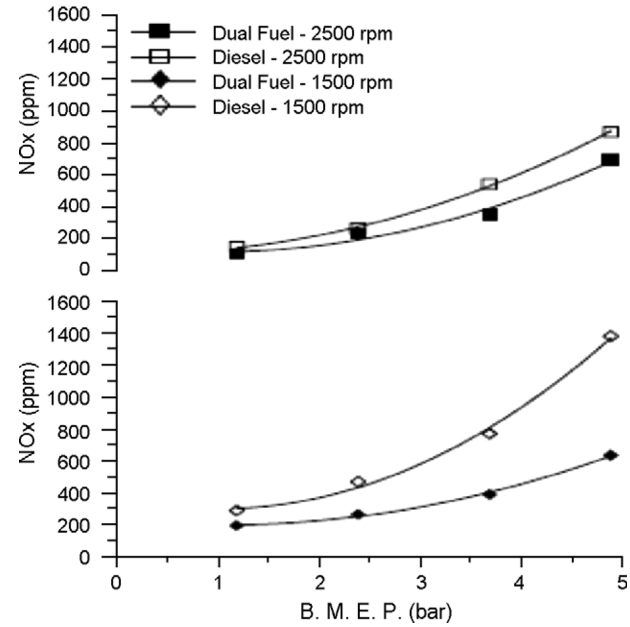

Fig. 10. Variation of nitric oxide under normal diesel and dual-fuel operation versus load at 1500 and 2500 rpm engine speed [19].

dual-fuel operation during the premixed controlled combustion phase is significantly lower compared to the one under normal diesel operation. It is revealed that the total rate of heat release under dual-fuel operation is obviously higher compared to the one under normal diesel operation (Fig. 7). The effect is stronger at low engine speed, revealing later combustion of the gaseous fuel and this obviously has an effect on the 'bsfc'.

The combustion duration is higher under dual-fuel operation at low engine load but drops with the increase of load (Fig. 8). Especially, at low engine loads, the combustion duration, even tends to become lower compared to normal diesel operation. At low engine loads, the total 'bsfc' for dual-fuel operation is considerably higher compared to the one under normal diesel operation (Fig. 9). This reveals a poor utilization of the gaseous fuel, mainly, due to the lower temperature and air–fuel ratio inside the combustion chamber of the engines, resulting in a slower combustion rate as observed from the results of the heat release rate analysis [11,20]. On the other hand, at high load, the improvement of gaseous fuel utilization leads to a relevant improvement of the total 'bsfc' under dual-fuel operation, which tends to converge to the one under normal diesel operation. But its value continues to be higher compared to the one under normal diesel operation.

The formation of nitric oxide (NO) is favored by high oxygen concentration and high charge temperature [11,21,22]. NO_x concentration is affected considerably by the presence of gaseous fuel–air mixture. The concentration of NO_x under dual-fuel operation is lower compared to the one under normal diesel

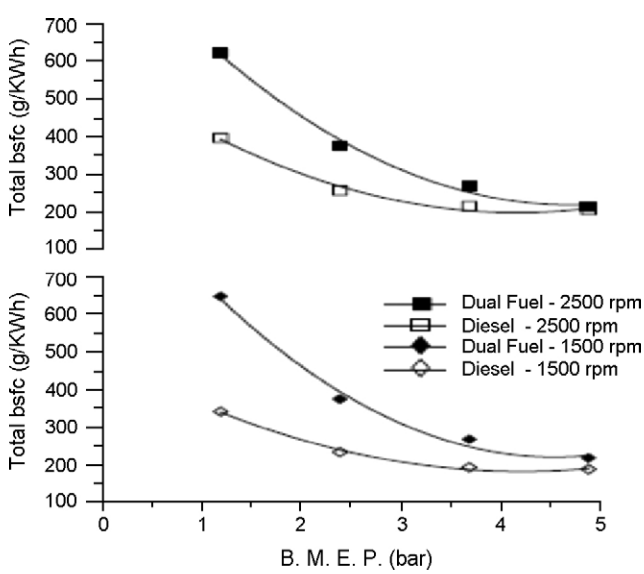

Fig. 9. Variation of total 'bsfc' under normal diesel and dual-fuel operation as function of load at 1500 and 2500 rpm engine speed [19].

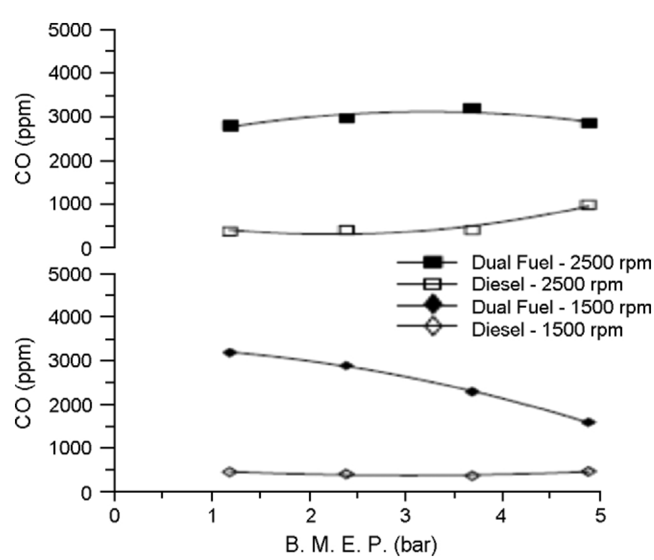

Fig. 11. CO under normal diesel and dual-fuel operation versus load at 1500 and 2500 rpm engine speed [19].

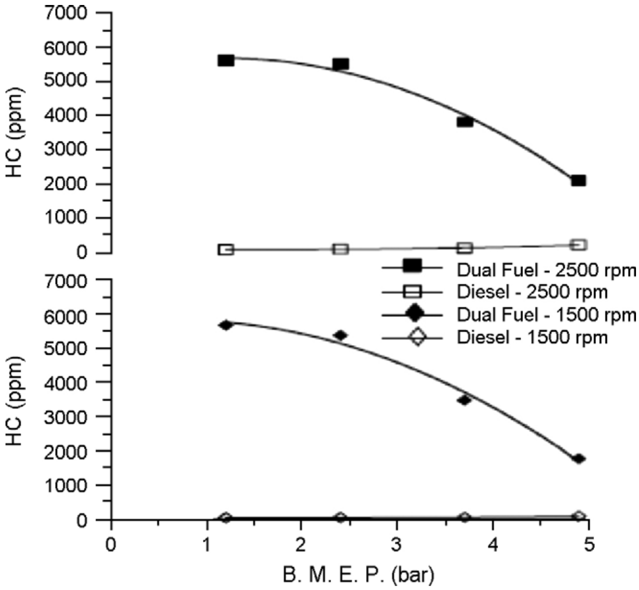

Fig. 12. UBHC under normal diesel and dual-fuel operation versus load at 1500 and 2500 rpm engine speed [19].

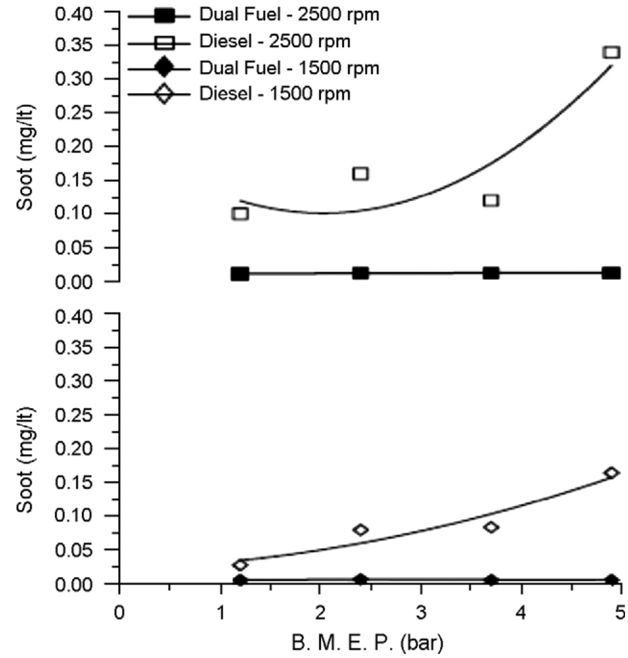

Fig. 13. Soot emissions under normal diesel and dual-fuel operation versus load at 1500 and 2500 rpm engine speed [19].

operation at the same engine operating conditions (i.e. engine speed and load). At low engine loads, the NO_x concentration under dual-fuel operation is slightly lower compared to the one under normal diesel operation (Fig. 10). This is mainly as a result of the lower rate of premixed controlled combustion of the gaseous fuel, which results in lower charge temperature inside the combustion chamber compared to normal diesel operation. At higher load, the NO_x concentration under dual-fuel operation is considerably lower compared to the one under normal diesel operation (Fig. 10). As far as the effect of engine speed is concerned, the increase of engine speed under dual-fuel operation results in a further decrease of NO_x values compared to normal diesel operation. The rate of CO formation is a function of the unburned gaseous fuel availability and mixture temperature which controls the rate of fuel decomposition and oxidation [11,21,23]. The CO emissions under dual-fuel operation are significantly higher as shown in Fig. 11. At low engine speed, CO concentration under dual-fuel operation clearly decreases with the increase of engine load. This is the result of the improvement of gaseous fuel utilization especially during the second phase of combustion. At high engine speed, the increase of engine load does not seem to affect the concentration of CO due to the less time available for combustion.

At low engine load, HC emissions under dual-fuel operation are considerably higher compared to the ones under normal diesel operation (Fig. 12). This is mainly due to the lower charge temperature and air–fuel ratio, resulting in slower combustion and allowing small quantities of fuel to escape the combustion process. With the increase of engine load, there is a sharp decrease of HC emissions under dual-fuel operation. This is the result of the increase of burned gas temperature that helps oxidize efficiently the UBHCs. But for all cases examined, the HC emissions are considerably higher under dual-fuel operation compared to normal diesel operation. From Fig. 13, it can be seen that soot emissions under dual-fuel operation are considerably lower compared to the ones under normal diesel operation for all cases examined. Under normal diesel operation, soot emissions increase with increasing engine load. On the other hand, under dual-fuel operation and for all cases examined, the soot emissions do not follow the same trend since a reduction of soot emissions is observed with increasing engine load. This is to be expected since the increase of engine load is accomplished by increasing the amount of gaseous fuel that forms no soot while the increasing charge temperature contributes to its oxidation [24–29].

Performance of DG set in dual-fuel mode (FD + producer gas) at different engine load conditions is investigated by Singh et al. [30]. Performance of DG is evaluated in terms of SEC, brake thermal efficiency, engine output and sound pressure level. In operating CI engine on dual-fuel mode it is noted that there is a minor reduction in engine output about 1–2% (Table 4). This reduction is expected, because the calorific value of air–producer gas mixture is lower than that of air–liquid fuel vapour mixture. The engine performance is

Table 4
Effect of fuel and load on engine output, SEC, brake thermal efficiency and sound pressure [30].

Engine load (%)	63	63	84	84	98	98
Mode of operation	FD	Dual-fuel	FD	Dual-fuel	FD	Dual-fuel
RPM of engine	1487	1490	1493	1483	1486	1489
Engine output (kW)	14.08	14.00	19.34	18.93	22.59	22.00
LFCR (kg h^{-1})	3.586	1.336	4.517	1.445	5.370	4.183
SEC (MJkW h^{-1})	11.07	19.55	10.15	15.39	10.35	11.61
η_{bth} (%)	32.53	18.41	35.45	23.38	34.77	31
PGFR (m^3 h^{-1})	–	50	–	53	–	17
LFR (%)	–	62.74	–	68.00	–	22.24
Sound pressure (db)	100.5	96.90	101.50	102.15	99.50	100.40

FD, fossil-diesel; LFR, liquid fuel replacement (%); PGFR, producer gas flow rate (m^3 h^{-1}); LFCR, liquid fuel consumption rate (kg h^{-1}).

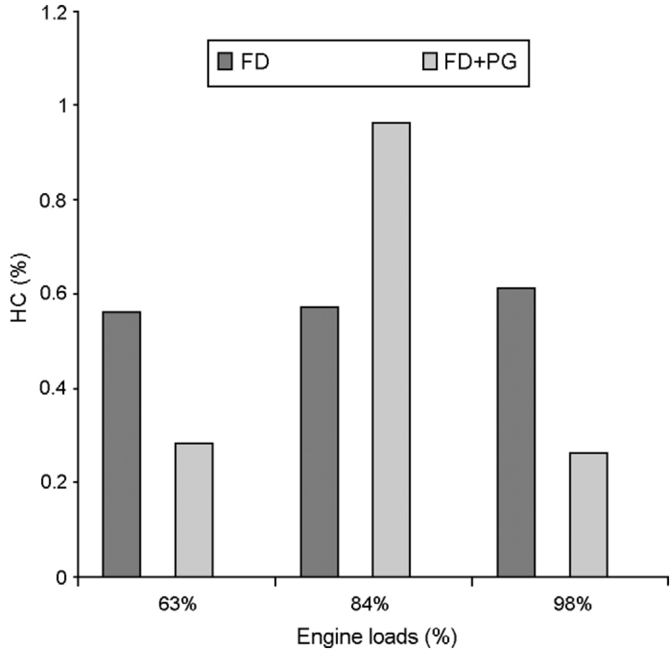

Fig. 14. Effect of fuels and engine loads on HC [30].

found to be extremely sensitive to the variation in the gas quality. When the engine operates with a maximum liquid fuel replacement the deterioration in the gas quality has an adverse effect on performance of the engine. This is due to the fact that during such operation the air intake is reduced to a level which is just enough for the combustion of liquid fuel and producer gas. As the quality of producer gas deteriorates with a consequent reduction in the engine speed, the governor is trying to keep the speed up, while allowing more liquid fuel in to the engine. This in turn causes incomplete combustion and misfiring due to insufficient combustion air. Even, the stalling of the engine will follow this if the air intake is not increased.

At 84% engine load with 18.4:1 CR in dual-fuel mode, the percentage reduction in concentration of pollutants like CO, CO_2, NO, NO_2 is 55.5%, 19.7%, 82% and 83%, respectively, while HC concentration increases by 67.2% (Figs. 14–18). The exhaust gas temperature increases when CR and index of compression are independently increased (Fig. 19).

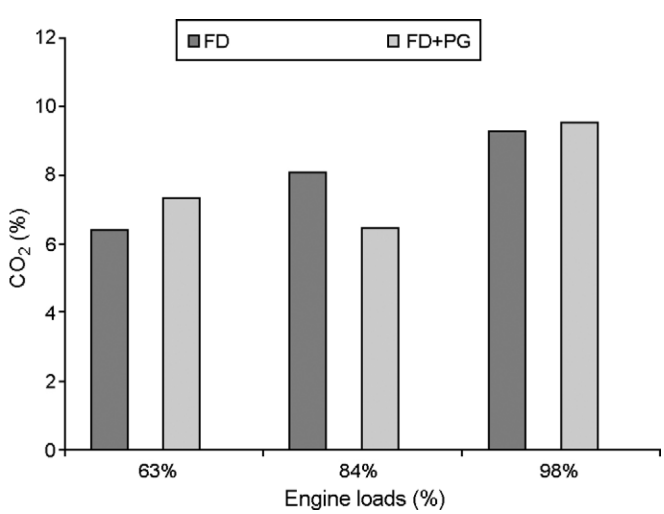

Fig. 15. Effect of fuels and engine loads on CO_2 [30].

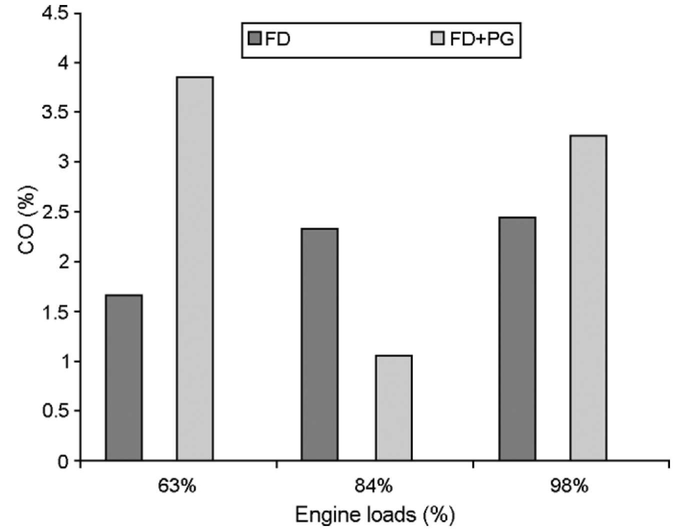

Fig. 16. Effect of fuels and engine loads on CO [30].

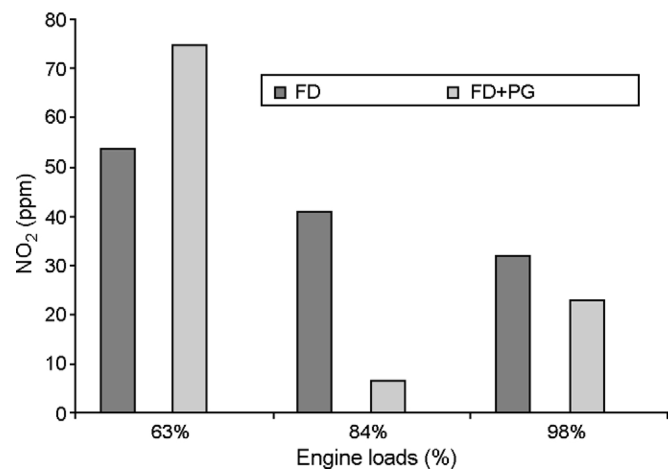

Fig. 17. Effect of fuels and engine loads on NO_2 [30].

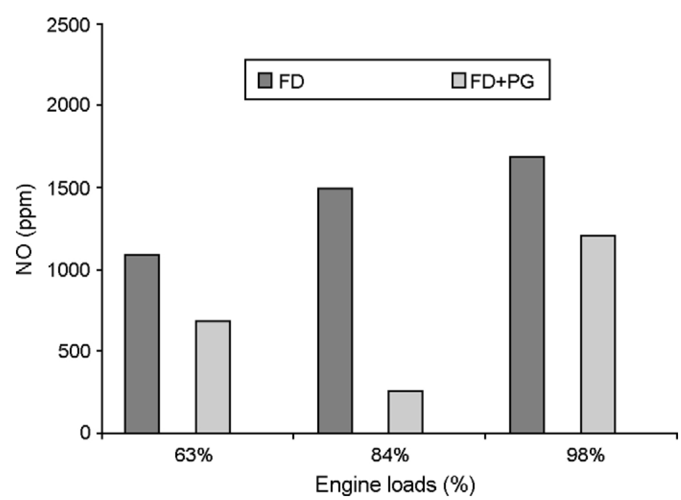

Fig. 18. Effect of fuels and engine loads on NO [30].

5.2. Effect of engine speed

The emission and performance levels are measured on a chassis dynamometer of a diesel engine under steady and unsteady

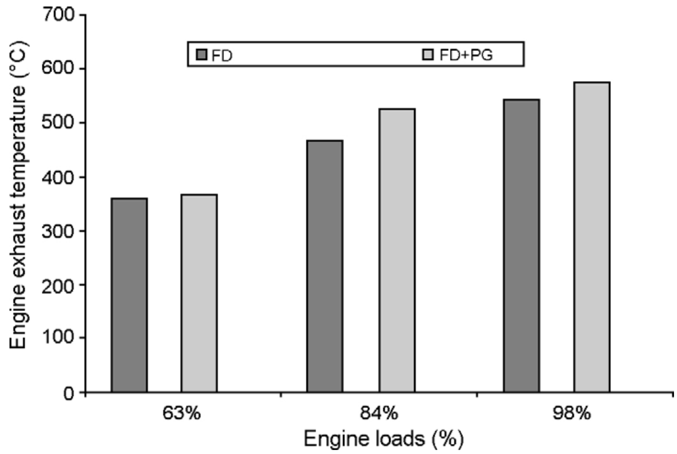

Fig. 19. Effect of fuels and engine loads on engine exhaust temperature [30].

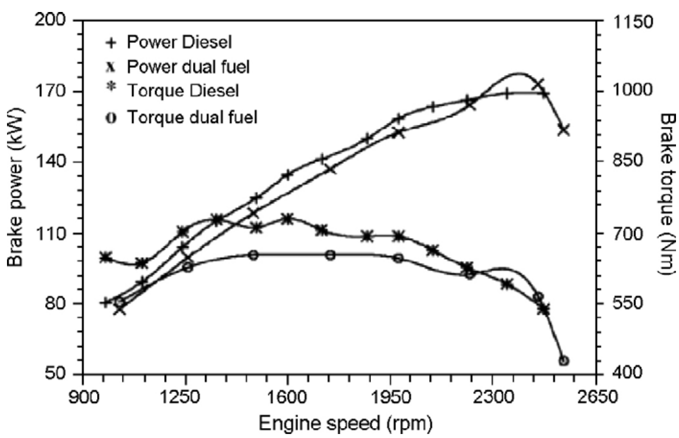

Fig. 20. Brake power and brake torque as a function of engine speed for full load [12].

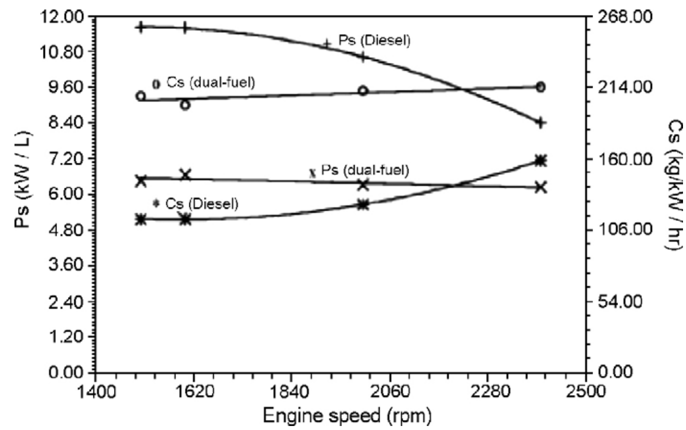

Fig. 21. 'bsfc' and brake specific power as a function of engine speed for full load [12].

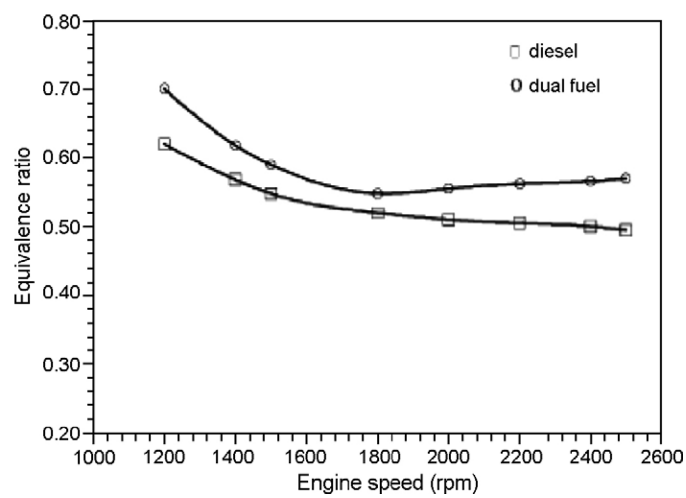

Fig. 22. Fuel–air equivalence ratio versus engine speed for full load [12].

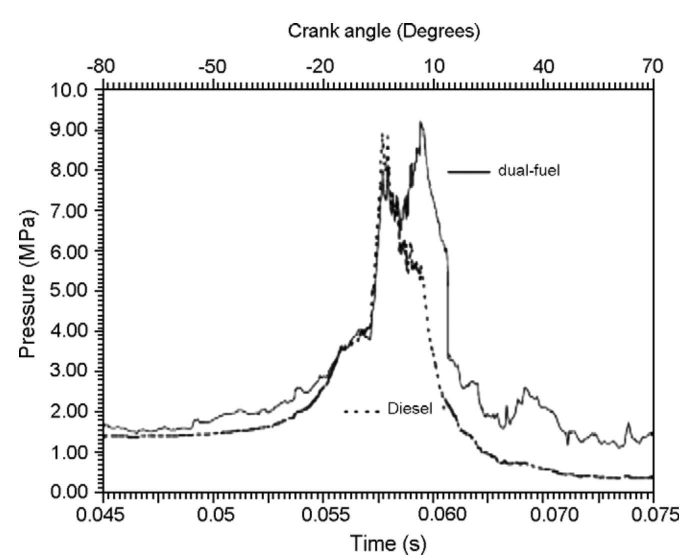

Fig. 23. Cylinder pressure data for diesel and dual-fuel gas fuelling for full loads at 1000 rpm engine speed [12].

conditions by Mansour et al. [12]. The compression ratio is kept constant for the two versions engine test. Under unsteady conditions, a small power and torque losses in dual-fuel version is observed as compared to the diesel one, except around the speed of 2400 rpm (Fig. 20). This difference is due to the system response time of gaseous fuel injection in the admission collector. When the engine speed rises to 2450 rpm, the brake torque and the brake power in dual-fuel version becomes slightly higher than in the diesel mode. Fig. 21 shows the evolution of the specific power and fuel consumption versus engine speed for full load. It is noticed that at low engine speed the difference in brake torque and power is important between the two versions (loss of power and higher consumption), but in high regimes the gap becomes less important. In this analysis, the evaluation of the specific consumption is based on the experimental results of the brake power and the consumption results under steady conditions.

The NG fuelling compared to diesel fuelling leads to slightly higher (less lean) equivalence ratios for a given speed condition (Fig. 22). This occurs for two reasons; first, the NG is aspirated into the engine where it mixes with air, thereby displacing some portion of air, which could have moved into the cylinder. As less air is induced, the equivalence ratio increases. Secondly, load is decreased, the engine is less efficient using NG, and hence more NG must be added to produce the fixed load-speed condition. The increased fuelling then increases the equivalence ratio of the engine. The cylinder pressure data measured with NG fuelling show (Figs. 23 and 24) the second pressure peak occurring between 7° and 15° crank angles (depending on engine load and speed) after the corresponding pressure peak for diesel fuelling. The double hump showed on the pressure trace, is the combustion of the pilot diesel charge followed by the NG combustion. The

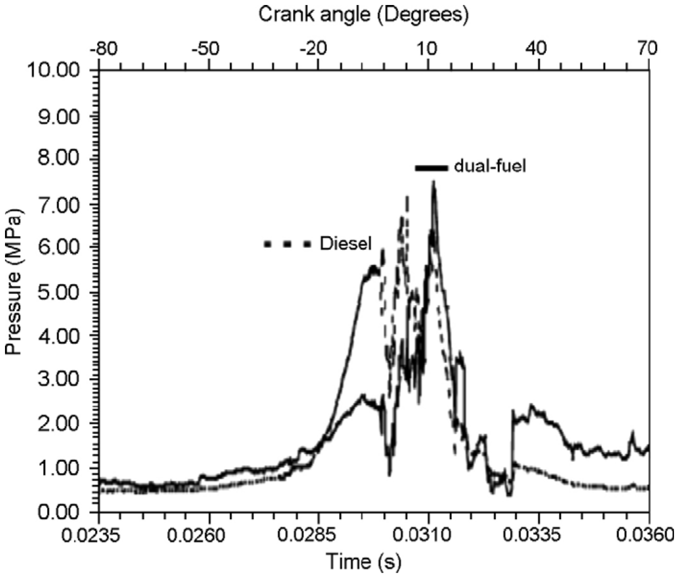

Fig. 24. Cylinder pressure data for diesel and dual-fuel gas fuelling for full loads at 2000 rpm engine speed [12].

maximum combustion pressure for NG fuelling is slightly higher for all engine speeds than the diesel fuelling level. The general trend is governed by decreasing pressure and temperature levels with increasing speeds.

Selim [14] has examined the effect of engine speed on combustion noise for diesel and dual-fuel cases (Fig. 25). The maximum pressure rise rate during combustion is presented as a measure of combustion noise. Generally, as the engine speed increases, the pressure rise rate ($dP/d\theta$) decreases for both mode of operation. In the diesel engine case, the pressure rise rate drops from about 5.5 bar/°C A at speed of 990 rpm to around 2.93 bar/°C A at 1890 rpm. However, for the dual-fuel case the pressure rise rate is higher than that for the diesel case, almost at all engine speeds, and it follows a similar trend to the diesel mode. It drops from 6.6 bar/°C A at 980 rpm to 2.95 bar/°C A at 1880 rpm. It is also been shown in an IDI diesel engine [31] that the combustion noise decreased when the engine speed increases. Further, Selim [4] has presented the effects of engine speed for a dual-fuel engine (Fig. 26a–d). During these experiments, the constant parameters

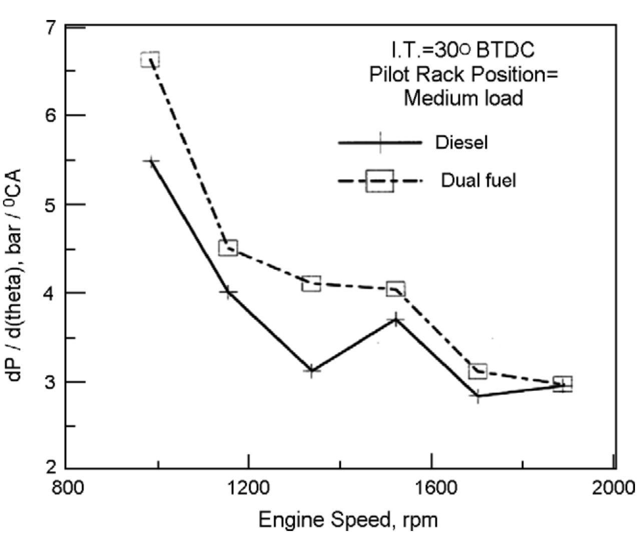

Fig. 25. Effect of engine speed on pressure rise rate for the diesel and dual-fuel engines [14].

are; pilot fuel injection timing 35° BTDC, mass of pilot fuel 0.37–0.47 kg/h and compression ratio 22. From Fig. 26(a) and (b), it is seen that the LPG produces the lowest torque output and thermal efficiency as compared to methane or the NG mixture. The torque output and efficiency is highest for methane gas. The thermal efficiency also improves with increasing engine speed. Fig. 26(c) and (d) shows that, the pressure rise rate ($dP/d\theta$) decreases as the engine speed increases for the three dual-fuel cases. However, the pressure rise rate is highest for the dual-fuel engine and follows the same trend with methane, followed by LPG and NG at almost all engine speeds.

5.3. Effect of pilot fuel injection timing

The injection timing of the pilot fuel is an important factor that influences the performance of dual-fuel engines. For a fixed total equivalence ratio, advancing the injection timing increase the peak cylinder pressure because more fuel is burned before TDC and the peak pressure moves closer to TDC. Retarding the injection timing decreases the peak cylinder pressure because more of the fuel burns after TDC. This is because, the pilot fuel combustion is delayed and thus, the temperature of the mixture is not enough to propagate the flame in the whole gaseous fuel–air mixture; and consequently, incomplete combustion of the gaseous fuel mixture takes place. The charge temperature increases with advancing the injection timing of the pilot fuel and the associated higher energy release rates of the mixture. Similarly, the rates of pressure rise during the combustion of the gaseous fuel increases with advancing the injection timing of the pilot fuel.

The effect of advanced injection timing on the performance of NG used as primary fuel in dual-fuel combustion has been examined by Nwafor [2]. The injection is first advanced by 5.5° (i.e. 35.5° BTDC). The engine runs for about 5 min at this timing and then stops and with subsequent attempts, he fails to start the engine. But after changing the injection to 33.5° BTDC, the engine runs smoothly, but seems to incur penalty on fuel consumption especially at high load levels. Table 5 compares fuel consumption measurements for the two speeds, i.e. 3000 and 2400 rev/min of investigation. The data show that the advanced timing system incurs penalty on fuel consumption for a given operating condition. The governor injects more pilot fuel than when running on standard time units. The poor performance of gas engines at low load levels is due to the effect of gas residuals and low cylinder temperature. It is also due to the reduction in combustion efficiency caused by reduced flame propagation speed and increased compression work resulting from the large amount of air–gas inducted. The diesel fuel operation produces the highest BTE at the two speeds (Fig. 27a and b). Standard timing shows little improvement over the advanced system at 3000 rev/min. However, at 2400 rev/min the dual-fuel mode standard and advanced timing show similar trends at low and intermediate load levels. Fig. 28(a) and (b) shows that diesel fuel operation produces the shortest delay periods at both test speeds. Standard dual-fuel timing at 3000 rev/min shows longer delay periods at high loads than the advanced injection timing operation. There is a very significant difference between the ignition delays of diesel fuel and dual-fuel operations at 2400 rev/min. Standard dual-fuel timing also shows the longest delay period. At high loads and combustion temperatures, the ignition delay is reduced and combustion is dominated by the system temperature. At low loads with longer delay periods, greater proportion of pilot fuel takes part in premixed combustion, thus increases the tendency of diesel to knock. Very poor atomization results in a relatively long delay period, due to the slow development of very fine droplets. Dual-fuel operation with advanced timing shows the highest exhaust

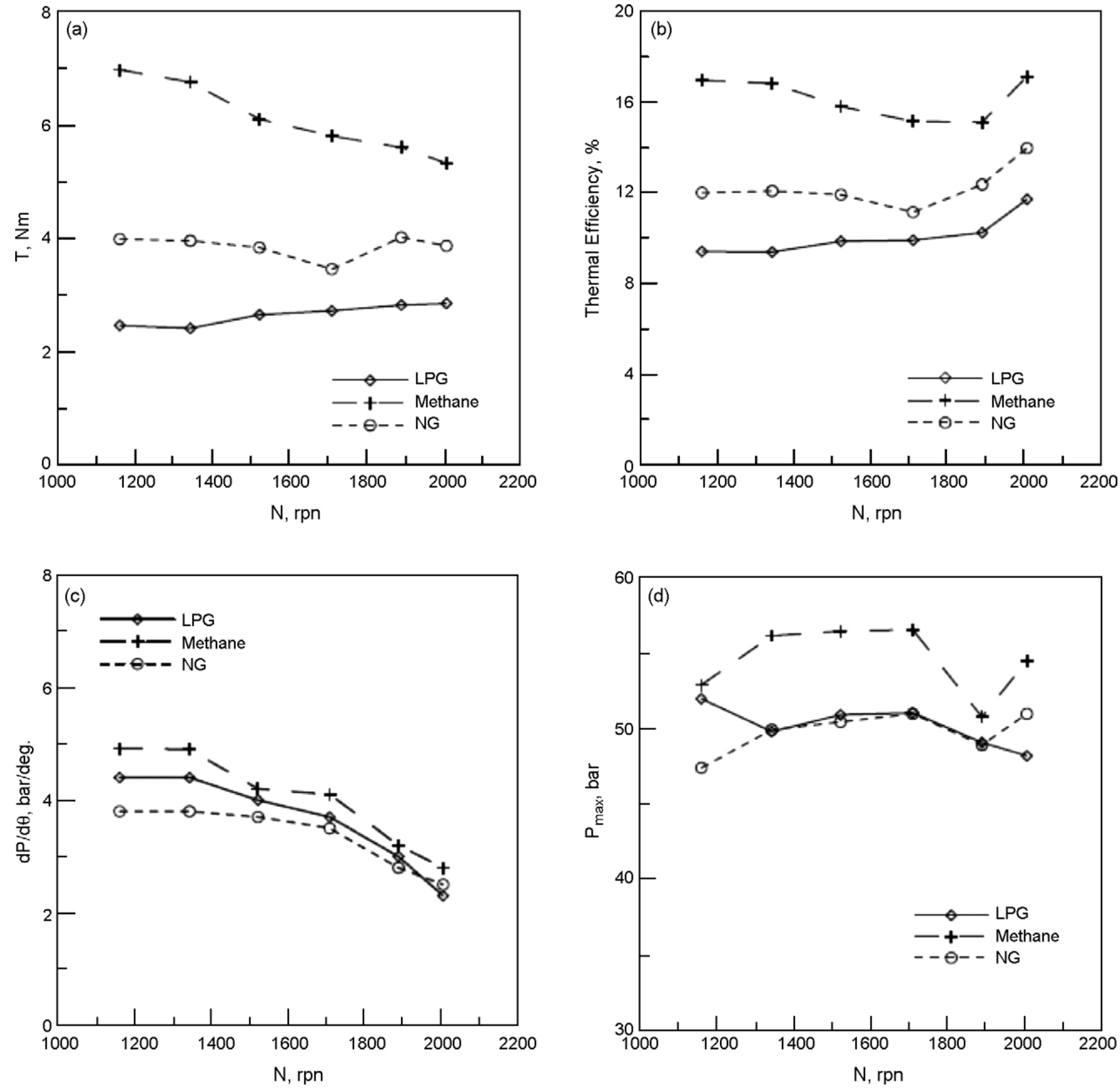

Fig. 26. Effects of engine speed on performance and noise: IT = 35° BTDC, CR = 22, m_d = (0.37–0.47)kg/h [4].

temperatures at 3000 rev/min (Fig. 29a and b). The dual-fuel mode with the standard timing also shows a marginal increase over the operation on pure diesel fuel. The standard timing dual-fuel unit produced the highest cylinder wall temperatures while the advanced system showed the lowest values at this speed. However, the diesel fuel operation produces the lowest cylinder wall temperatures, whereas the standard unit offers the highest values. The best fuel economy is realised when running on pure diesel fuel and hence, the thermal efficiency of the gas engine is less than that of pure diesel fuel case. The results from Fig. 30(a) and (b) indicate that the HC emissions of the gas-fuelled engine are higher than that in pure diesel fuel operation. Diesel fuel operation gives the lowest HC emissions at both speeds. Dual-fuel standard timing shows higher concentration of HC in the exhaust at low load levels over the advanced injection unit. However, this system shows decreased HC emission levels at high load level operation. HC emissions increase due to several factors, including quenching, lean combustion, wall wetting, cold starting and poor mixture preparation. For both test conditions, the HC levels are relatively high in dual-fuel operations and stay reasonably high throughout the load range.

Selim [14] has compared the effect of pilot diesel injection timing on the combustion noise (maximum pressure rise rate during combustion) of a dual-fuel engine to 100% diesel case is shown in Fig. 31. For the late injection of pilot (20–25° BTDC), the combustion noise is comparable for diesel and dual-fuel cases. However, when the injection advance increases (25–40° BTDC), the dual-fuel engine produced a higher rate of pressure rise ($dP/d\theta$). With the presence of gaseous fuel in the mixture, any advance in pilot fuel injection results longer ignition delay period and increase in pressure rise rate. Similar results are also shown by same author for LPG fuel [32].

At lower loads, dual-fuel engines suffer from lower thermal efficiency and higher unburned percentages of fuel. In order to

Table 5
Effect of advanced injection timing on the performance of NG [2].

Engine load (N)	Diesel fuel operation (kg/h)	Standard timing, pilot fuel consumed (kg/h)	Standard timing, gas consumed (l/min)	Advanced timing, pilot fuel consumed (kg/min)	Advanced timing, gas consumed (l/min)
Engine speed = 3000 rev/min					
6.61	0.539	0.368	10.0	0.408	10.0
13.18	0.642	0.402	11.0	0.456	11.0
17.72	0.696	0.442	12.4	0.478	12.5
23.78	0.818	0.447	14.0	0.496	14.0
29.33	0.894	0.470	16.0	0.509	16.0
33.87	0.975	0.493	17.0	0.535	17.0
38.92	1.045	0.529	18.0	0.594	18.0
44.47	1.154	0.563	19.0	0.642	19.0
49.52	1.266	0.570	20.2	0.631	20.2
Engine speed = 2400 rev/min					
8.63	0.463	0.333	8.0	0.349	8.0
13.68	0.509	0.299	12.0	0.319	12.0
18.73	0.592	0.278	14.0	0.298	14.0
24.79	0.680	0.279	15.2	0.295	15.2
29.83	0.735	0.303	16.0	0.324	16.0
35.89	0.829	0.312	17.0	0.366	17.0
41.44	0.887	0.328	17.5	0.383	17.5
46.80	1.013	0.344	18.0	0.456	18.0
53.56	1.132	–	–	–	–

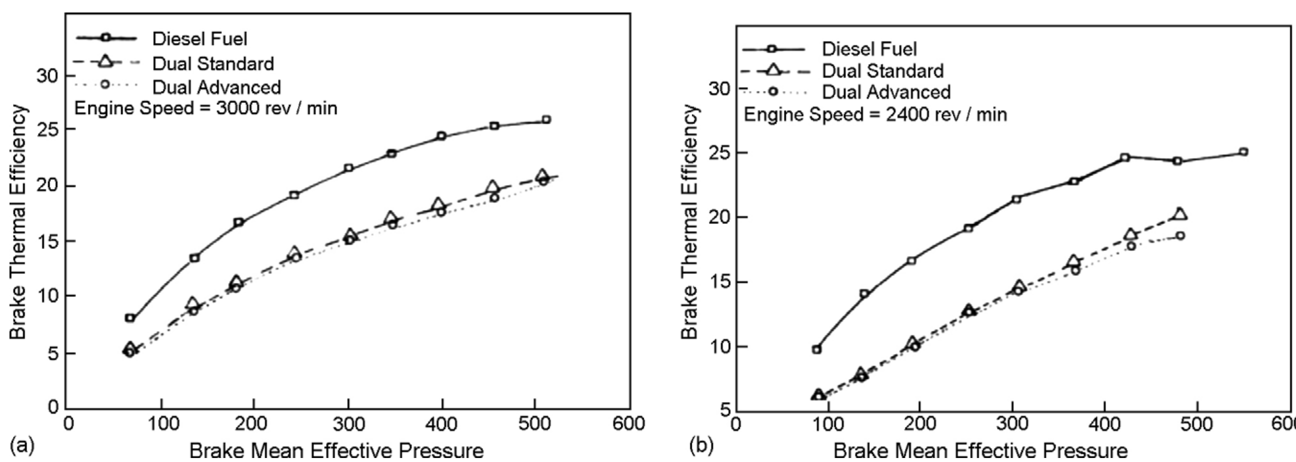

Fig. 27. Injection advanced effect on gas combustion—BMEP (kN/m^2) versus BTE (%) [2].

improve this, the effects of injection timings of 25°, 27.5° and 30° BTDC on the performance of an IDI diesel engine are investigated by Abd et al. [33]. Figs. 32 and 33 show that, advancing the pilot fuel injection timing reduces the UBHC emissions. This is due to a longer ignition delay of the mixture with the increased timing advance. The longer ignition delay allows a fuller spray penetration and development, creating a larger amount of the pilot fuel–air–gaseous fuel mixture (or flame propagation region) prior to

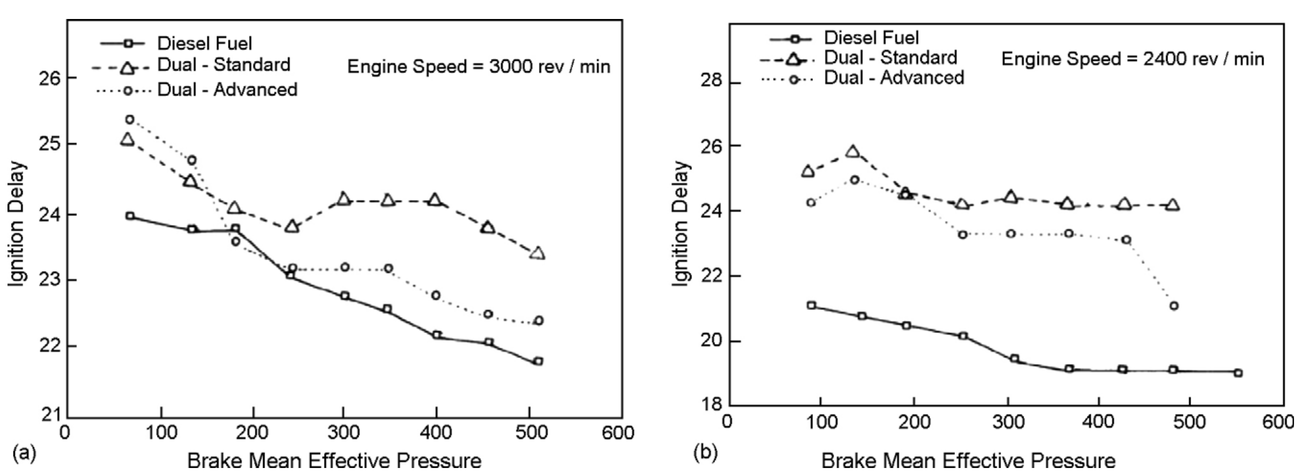

Fig. 28. Injection advanced effect on gas combustion—BMEP (kN/m^2) versus ignition delay (deg.) [2].

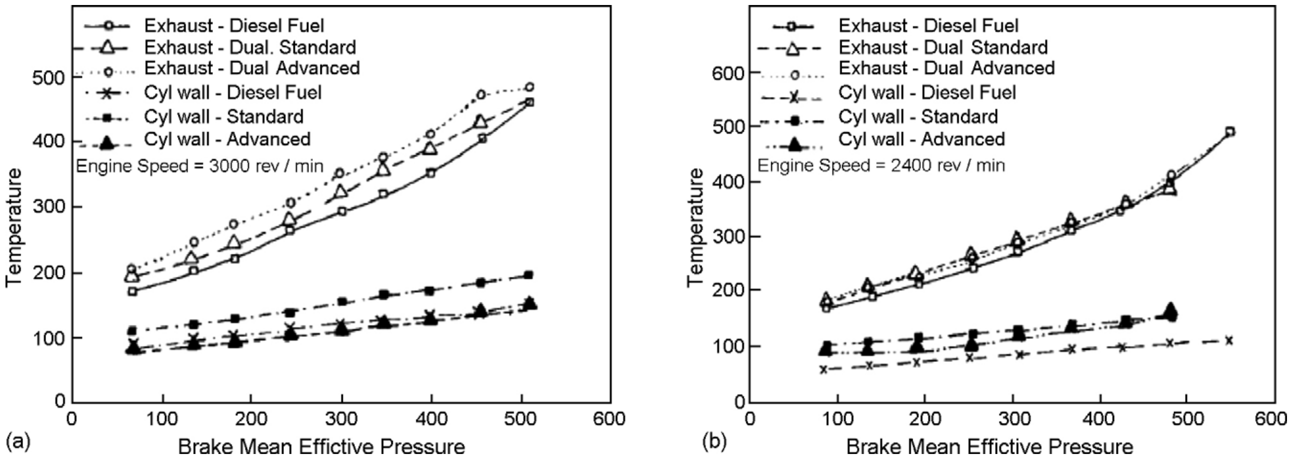

Fig. 29. Injection advanced effect on gas combustion—BMEP (kN/m^2) versus temperature (°C) [2].

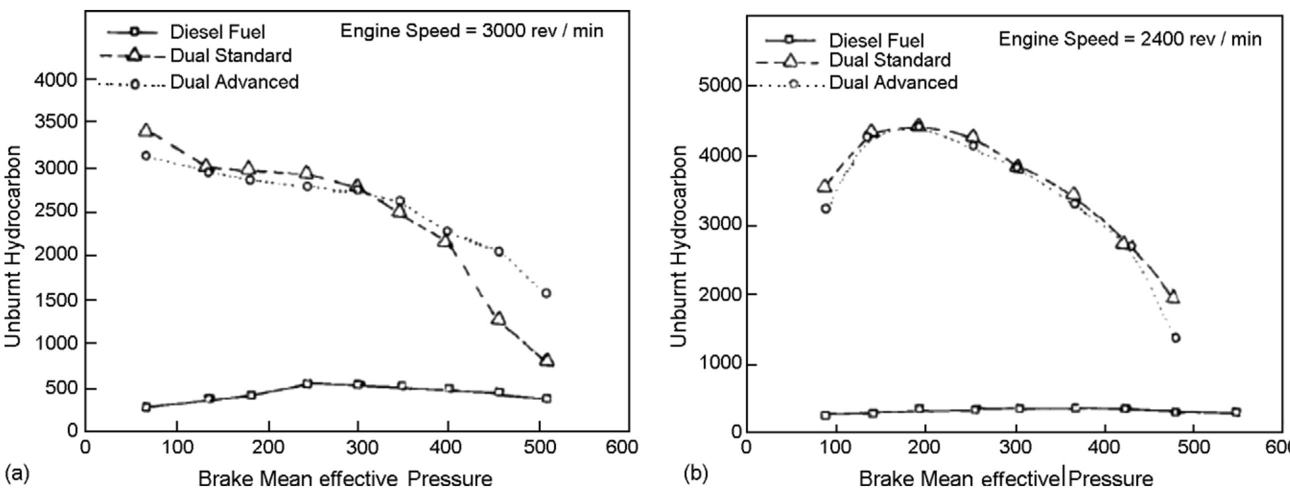

Fig. 30. Injection advanced effect on gas combustion—BMEP (kN/m^2) versus UBHC (ppm) [2].

ignition. The higher combustion rates of this larger premixed regions yields higher combustion temperatures and thus, lowers the UBHC emissions. With bigger injection advance, better overall combustion and the activity of the partial oxidation reactions reduce the CO emissions (Figs. 34 and 35). It also widens the lower combustion limit boundary of the overall lean mixture effectively.

For a fixed total equivalence ratio, amount of pilot fuel and intake temperature, advancing the injection timing has a great effect on the maximum charge temperature in the cylinder. For any total equivalence ratio, as the injection is retarded, the maximum charge temperature decreases. The net effect is a reduction in NO_x as shown in Figs. 36 and 37. Higher combustion temperatures due to

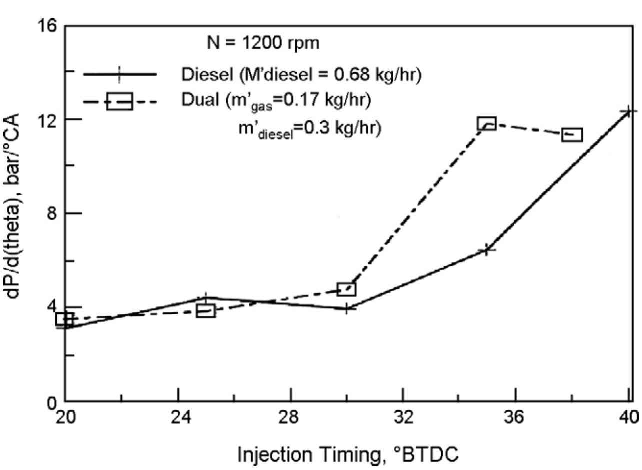

Fig. 31. Effect of pilot fuel injection timing on pressure rise rate for the diesel and dual-fuel engine [14].

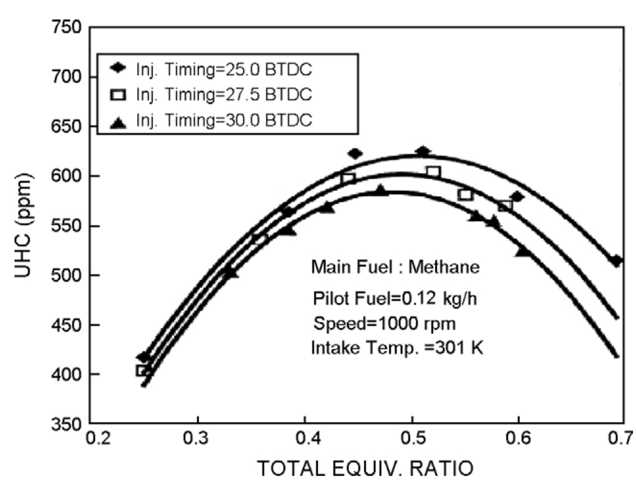

Fig. 32. Variations of experimental results of UBHC concentration with total equivalence ratio for different values of injection timings for methane [33].

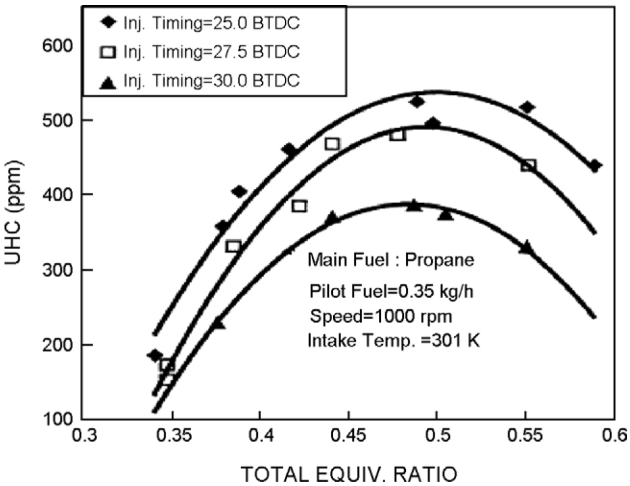

Fig. 33. Variations of experimental results of UBHC concentration with total equivalence ratio for different values of injection timings for propane [33].

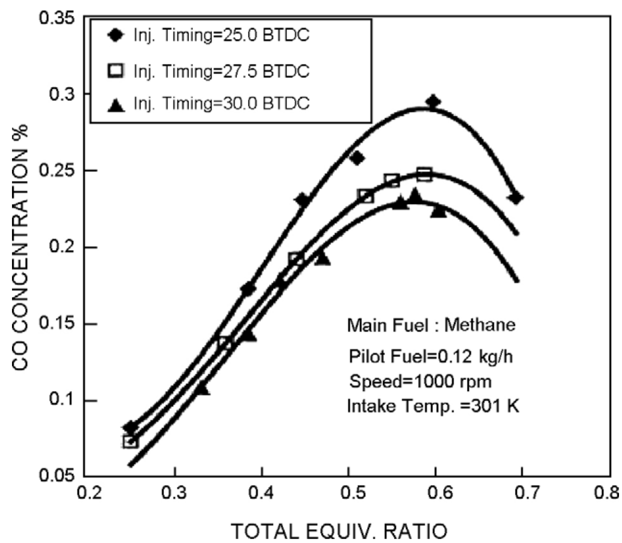

Fig. 34. Variations of experimental results of CO concentration with total equivalence ratio for different values of injection timings for methane [33].

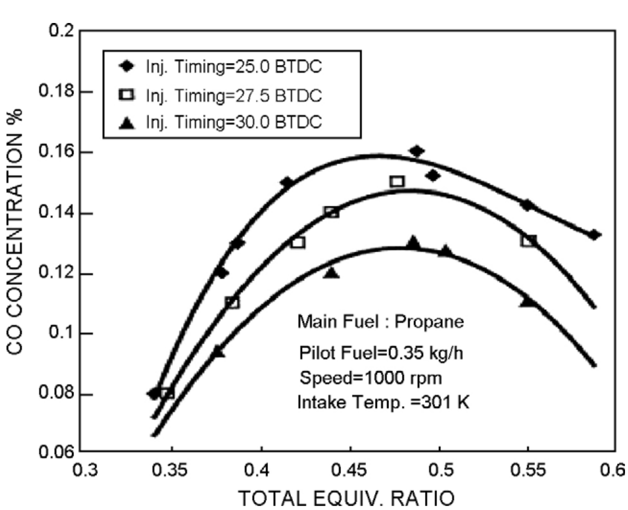

Fig. 35. Variations of experimental results of CO concentration with total equivalence ratio for different values of injection timings for propane [33].

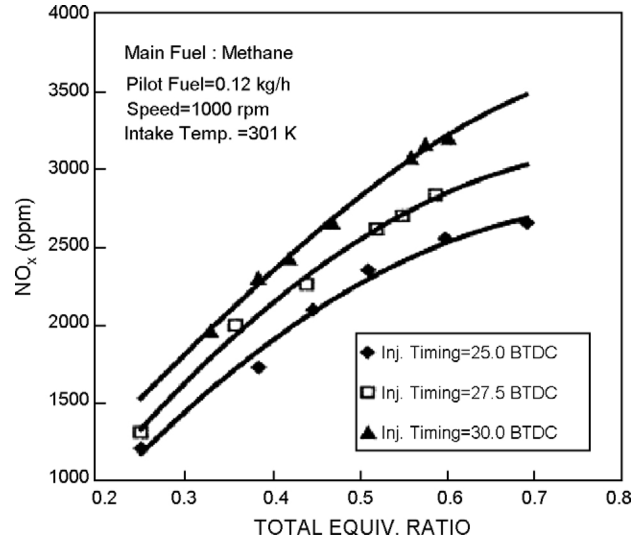

Fig. 36. Variations of experimental results of NO_x with total equivalence ratio for different values of injection timings for methane [33].

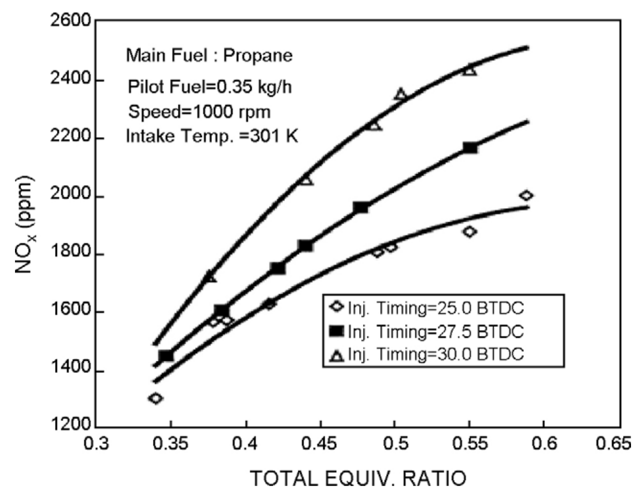

Fig. 37. Variations of experimental results of NO_x with total equivalence ratio for different values of injection timings for propane [33].

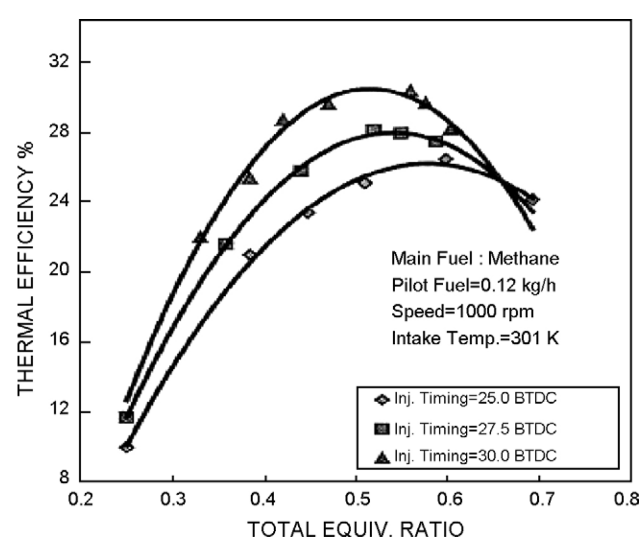

Fig. 38. Variations of experimental results of thermal efficiency with total equivalence ratio for different values of injection timings for methane [33].

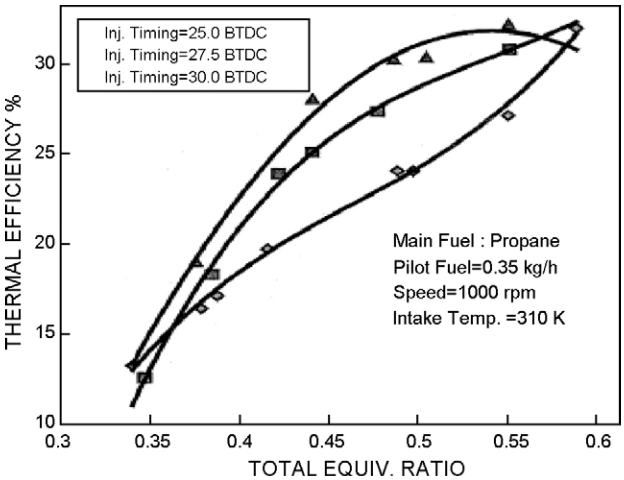

Fig. 39. Variations of experimental results of thermal efficiency with total equivalence ratio for different values of injection timings for propane [33].

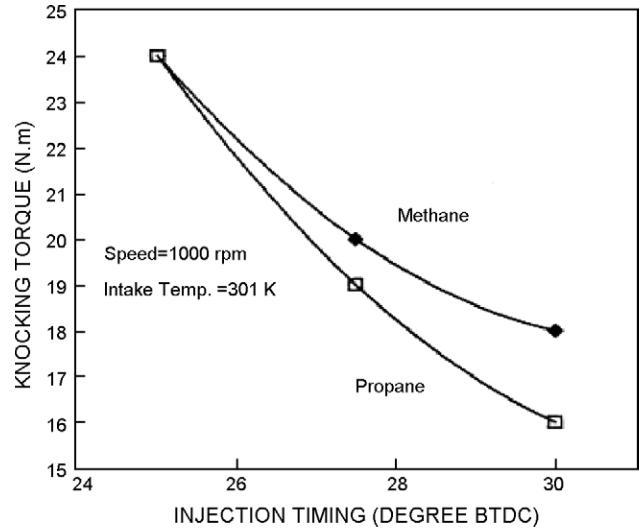

Fig. 40. Effect of injection timing on knocking torque for a dual-fuel engine fuelled with methane (CH_4) and propane (C_3H_8) [33].

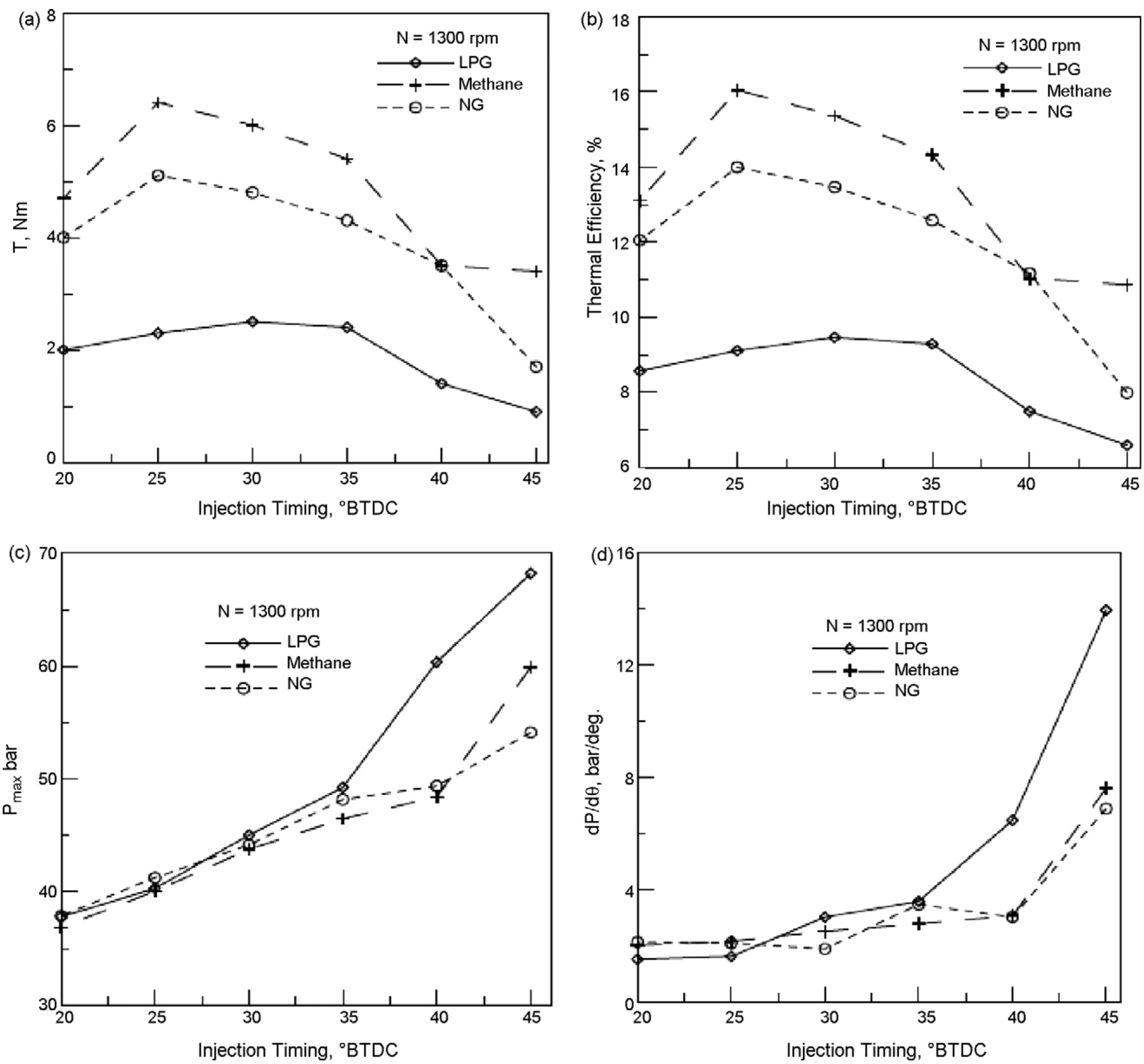

Fig. 41. Effects of pilot fuel injection timing on performance and noise: N = 1300 rpm, CR = 22, m_d = 0.37 kg/h [4].

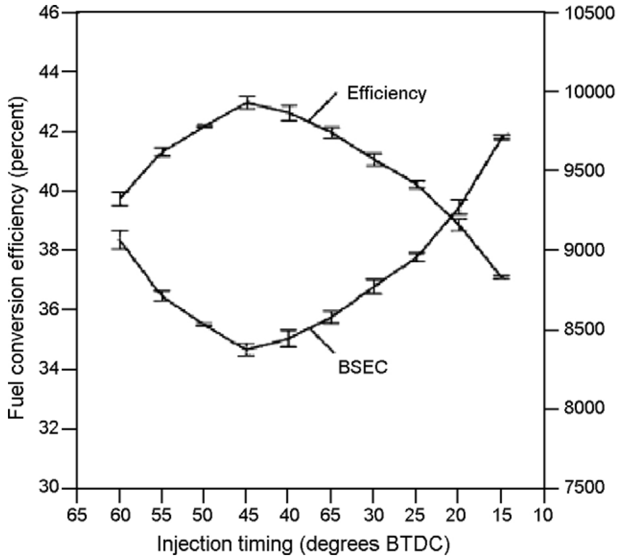

Fig. 42. Fuel conversion efficiency and BSEC versus injection timing [35].

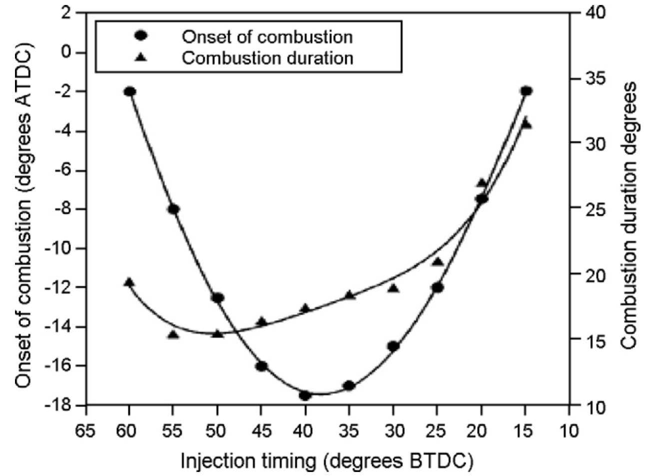

Fig. 44. Onset of combustion and combustion duration (10–90%) versus injection timing [35].

advancing the pilot fuel injection timing lead to increases in the brake power and, consequently, thermal efficiency (Figs. 38 and 39), respectively. Further advances in the injection timing will increase the tendency to knocking early for medium and high engine loads (Fig. 40).

Selim [4] has investigated the effects of pilot fuel injection timing for a dual-fuel diesel engine. The results are shown at constant speed of 1300 rpm and compression ratio of 22 (Fig. 41a–d). The highest torque output for methane and natural gas occurs when the injection timing is 25° BTDC, while for LPG it occurs at 30° BTDC. The torque output and hence the thermal efficiency is highest at certain timings, and it decreases at earlier or later timing (Fig. 41a and b). Earlier injection of pilot fuel causes the maximum pressure to increase, Fig. 41(c), and occurs BTDC in the compression stroke. This in turn, reduces the maximum pressure during the expansion stroke and torque output. The combustion noise, as shown in Fig. 41(d), increases as the pilot diesel injection advance increases for all dual-fuel cases. This is attributed to the increase in ignition delay of the diesel fuel, since the liquid fuel injected earlier in lower air pressure and temperature. The longer delay period would result in a higher pressure rise rate ($dP/d\theta$). This is also shown in by Abdalla et al. [34] for 100% diesel and all diesel–methanol blends.

To achieve low NO_x and good fuel efficiency from pilot-ignited natural gas combustion, Krishnan et al. [35] have tried different pilot injection timings in a single cylinder CI engine. The overall set of experiments involved engine testing at a constant speed of 1700 rev/min, full load (42 kW, 1220 kPa bmep) engine operation at fixed pilot quantity and inlet conditions. Fuel conversion efficiency (defined by the ratio of the brake power to the total rate of energy input into the engine) is calculated from the measured total diesel and natural gas flow rates multiplied by their respective lower heating values. Brake specific energy consumption (BSEC) is then calculated which is the reciprocal of the fuel conversion efficiency. As injection timing is advanced from 15° to 45° BTDC, fuel conversion efficiency increased from 38% to about 43% (Fig. 42). Upon further advance of the injection timing, the efficiency decreases to about 40% at 60° BTDC. When brake power and engine speed are held constant for all injection timings, the particular heat release pattern that provides for greater work per unit mass of fuel translates into higher fuel conversion efficiency. As injection timing is retarded from 35° to 15° BTDC, the start of

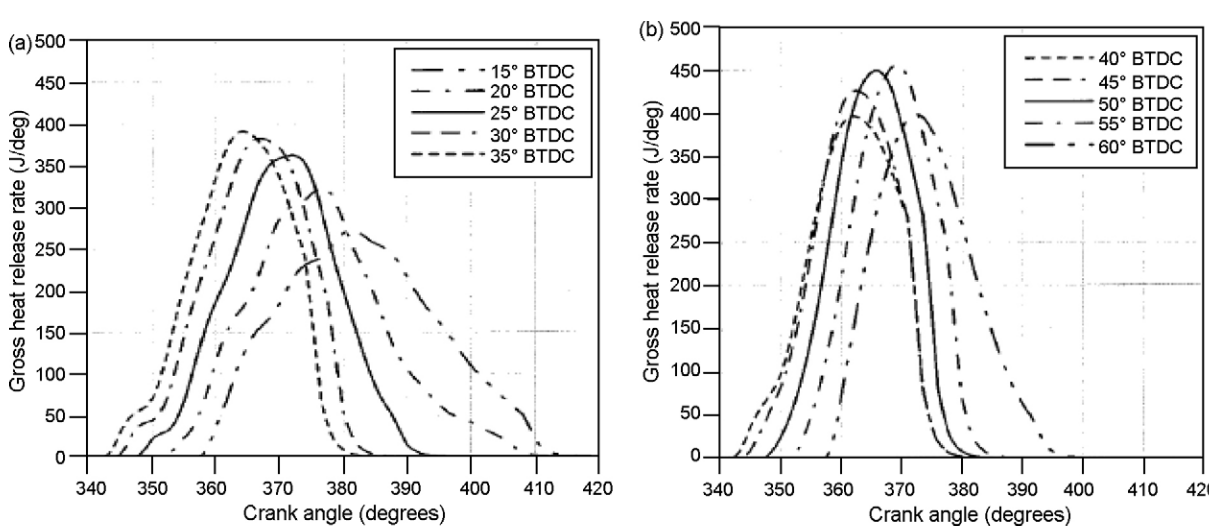

Fig. 43. (a) Gross heat release rate versus crank angle for injection timings between 15° and 35° TDC. (b) Gross heat release rate versus crank angle for injection timings between 40° and 60° BTDC [35].

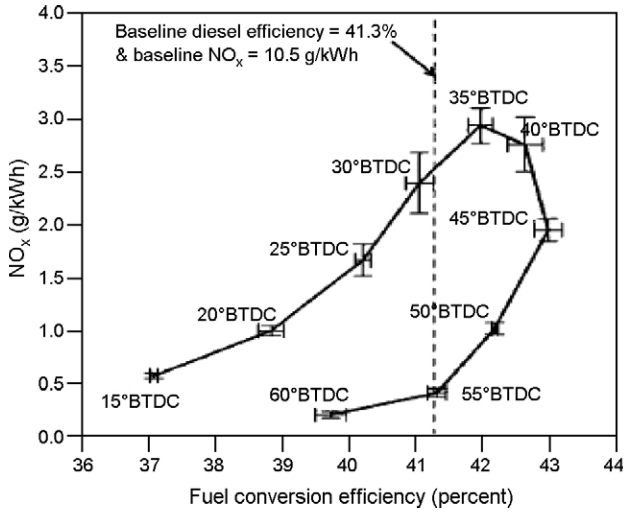

Fig. 45. NO_x-fuel conversion efficiency trade-off curve at different injection timings [35].

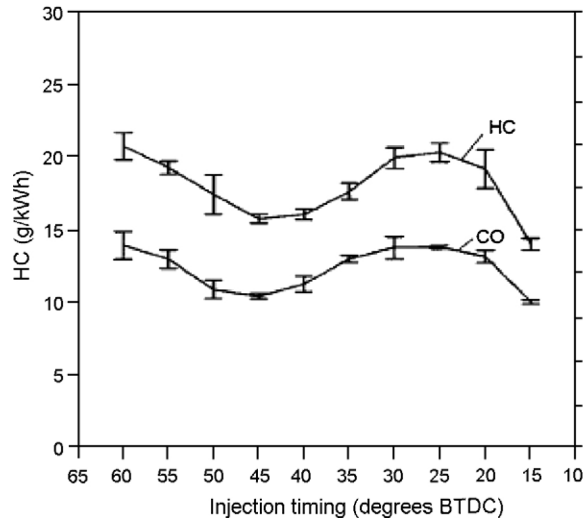

Fig. 46. HC and CO emissions versus injection timing [35].

heat release is delayed and the overall heat release process is shifted away from TDC. Also the duration and peak heat release are increased (Fig. 43a). These effects contributed to the loss in fuel conversion efficiency between 35° and 15° BTDC. As injection timing is advanced from 40° to 60° BTDC, the duration of heat release does not change significantly even though the start of heat release is delayed (Fig. 43b). The delayed start of heat release has led to a decrease in fuel conversion efficiency for timings advanced beyond 45° BTDC. Fig. 44 shows the onset of combustion in degrees ATDC and duration of combustion for the results shown in Fig. 43(a) and (b). The duration of combustion shown here is the period in crank angles from 10 to 90% mass burn. The onset of combustion is seen to be earlier for injection timings between 30° and 45° BTDC. For timings advanced beyond 45° BTDC or retarded beyond 30° BTDC, combustion begins progressively closer to TDC. However, combustion duration does not follow the same trend. For instance, although combustion has started at about the same time for both 15° and 60° BTDC, the combustion duration for the former is seen to be much longer than that of the latter. So, prolonged combustion time for 15° BTDC progresses into the expansion stroke, thereby reduces the fuel conversion efficiency compared to 60° BTDC. NO_x emissions for the maximum efficiency timing (45° BTDC) are much higher than those for very retarded (15° BTDC) or very advanced (60° BTDC) timings (Fig. 45). Along with NO_x reduction, efficiency also decreases when injection advances or retards from 45° BTDC. However, it is interesting that the efficiency for 60° BTDC is about 3% greater than that for 15° BTDC and NO_x emissions are lower as well. Thus, it is clearly more beneficial to advance the pilot injection timing to reduce NO_x emissions and maintain minimal loss in fuel conversion efficiency. Fig. 46 shows that for timings retarded beyond 25° BTDC, both HC and CO emissions are found to decrease especially between 20° and 15° BTDC timings. From Fig. 43(a) and 44, it is seen that for both 20° and 15° BTDC timings, the heat release peaks are lower and the combustion durations are longer compared to 25° BTDC. Although slower combustion have resulted in more wall-quenched HC for 20° and 15° BTDC, the time available for oxidation also increased since combustion duration increased. For the longest combustion duration (15° BTDC timing), the time for which high temperatures persist in the cylinder is increased, and hence more complete oxidation of HC and CO have occurred.

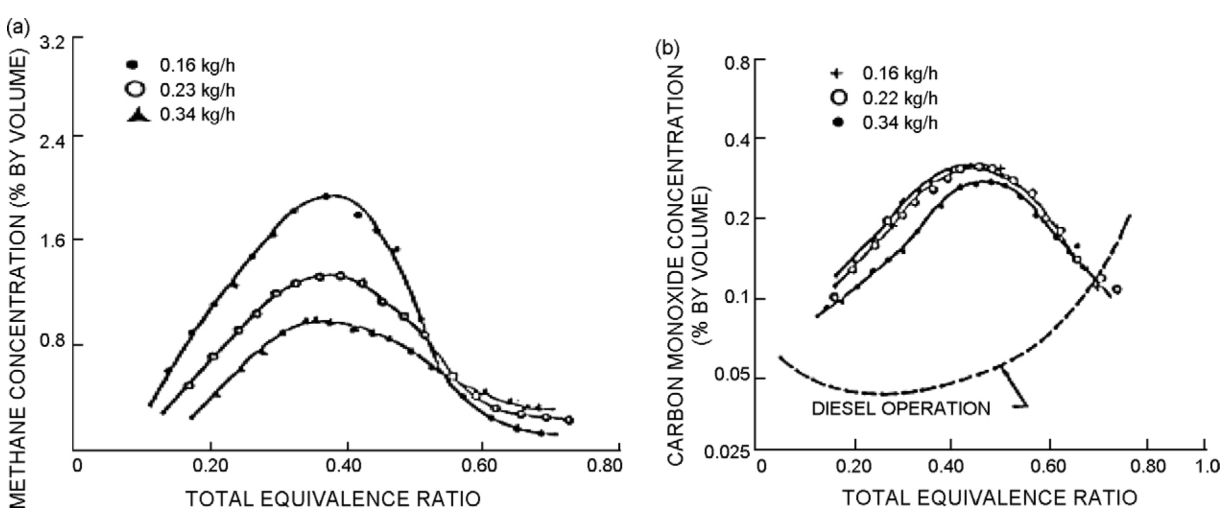

Fig. 47. The variations of the exhaust gas concentrations of methane and CO with total equivalence ratio for different pilot fuel quantities at ambient intake conditions and 1000 rpm [9].

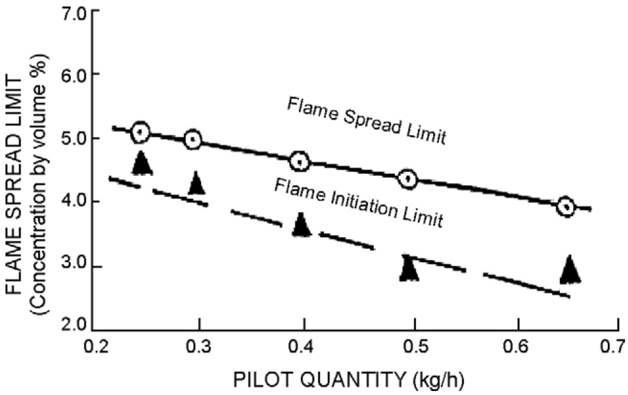

Fig. 48. Variations of the experimentally established flame spread limits (FSL) with the pilot quantity employed for methane operation at a compression ratio of 14.2:1 and 1000 rpm. The corresponding flame initiation limit values are also shown [9].

5.4. Effect of pilot fuel mass inducted

The pilot fuel quantity is one of the most important variables that have a controlling influence on the performance of dual-fuel engines, especially at light loads. It is known that most diesel fuel

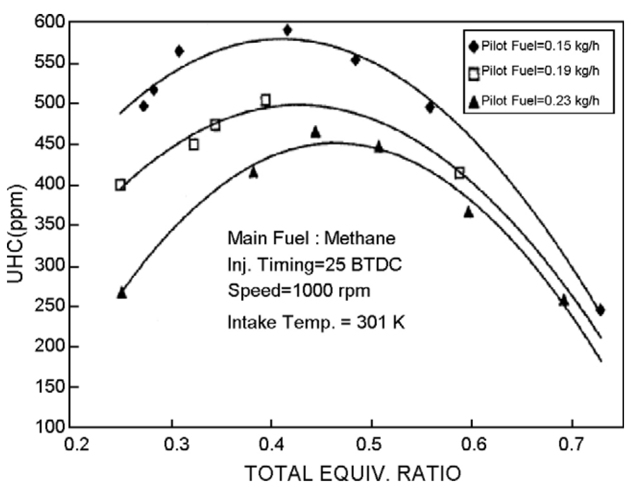

Fig. 49. Variations of experimental results of UBHC concentration with total equivalence ratio for different values of pilot fuel quantities (main fuel: methane) [36].

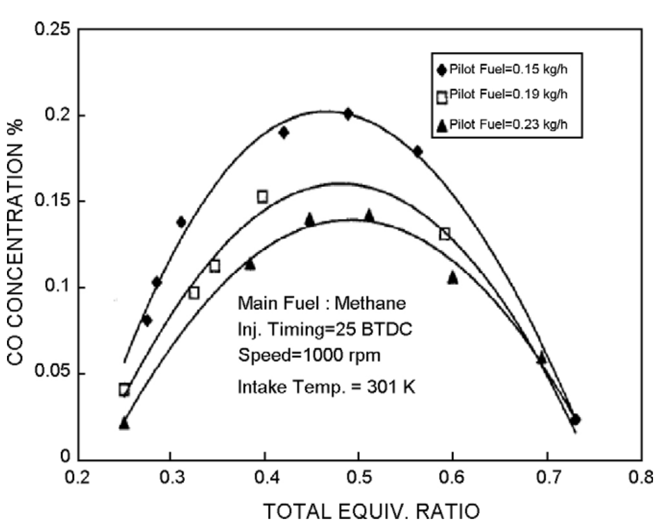

Fig. 50. Variations of experimental results of CO concentration with total equivalence ratio for different values of pilot fuel quantities (main fuel: methane) [36].

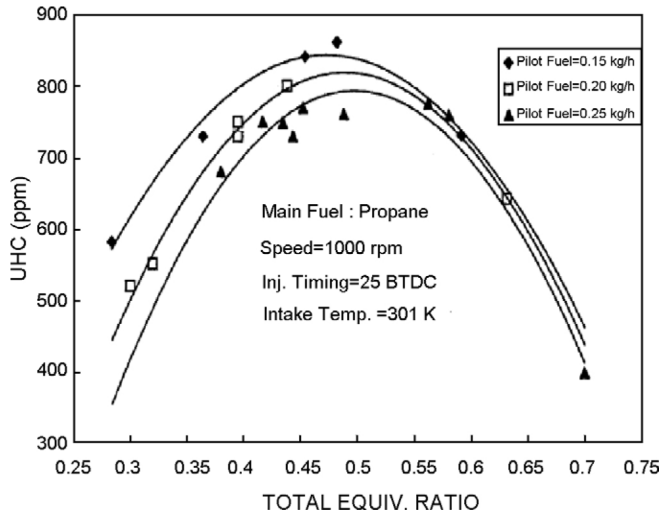

Fig. 51. Variations of experimental results of UBHC concentration with total equivalence ratio for different values of pilot fuel quantities (main fuel: propane) [36].

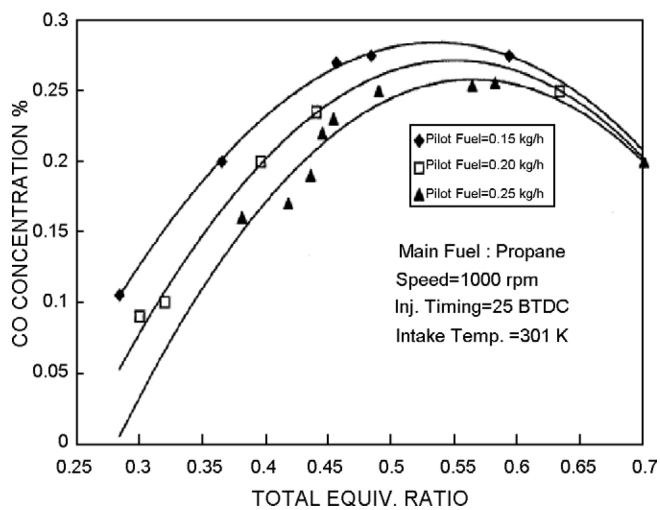

Fig. 52. Variations of experimental results of CO concentration with total equivalence ratio for different values of pilot fuel quantities (main fuel: propane) [36].

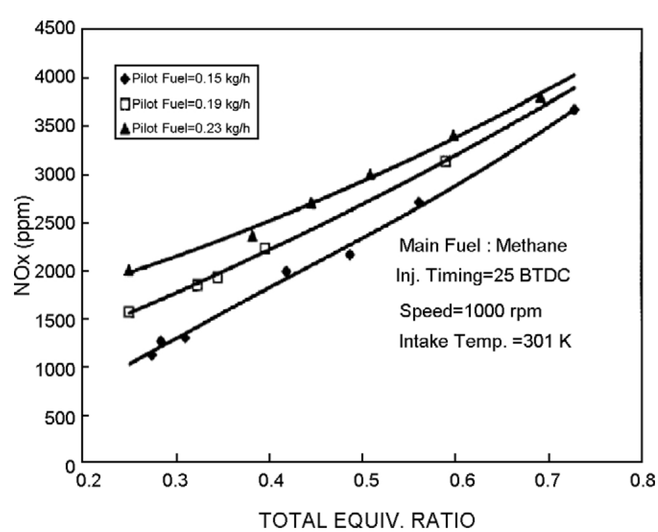

Fig. 53. Variations of experimental results of NO_x with total equivalence ratio for different values of pilot fuel quantities (main fuel: methane) [36].

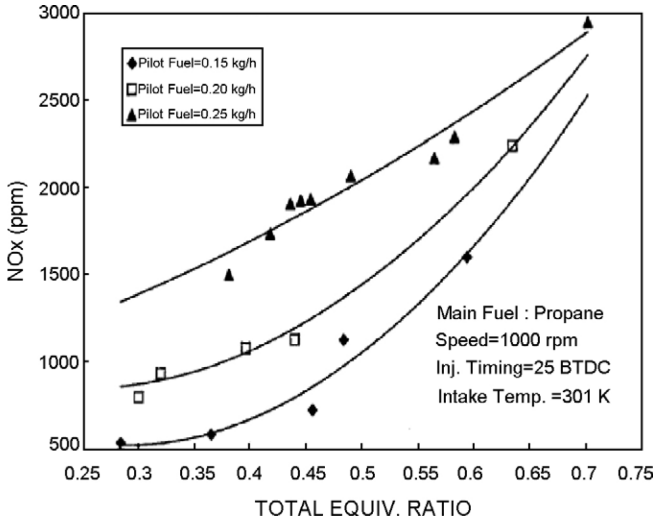

Fig. 54. Variations of experimental results of NO_x with total equivalence ratio for different values of pilot fuel quantities (main fuel: propane) [36].

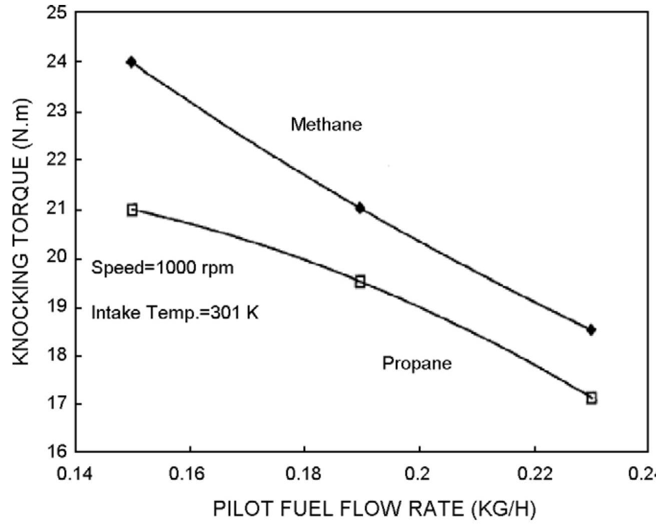

Fig. 57. Effect of pilot fuel flow rate on the knocking torque [36].

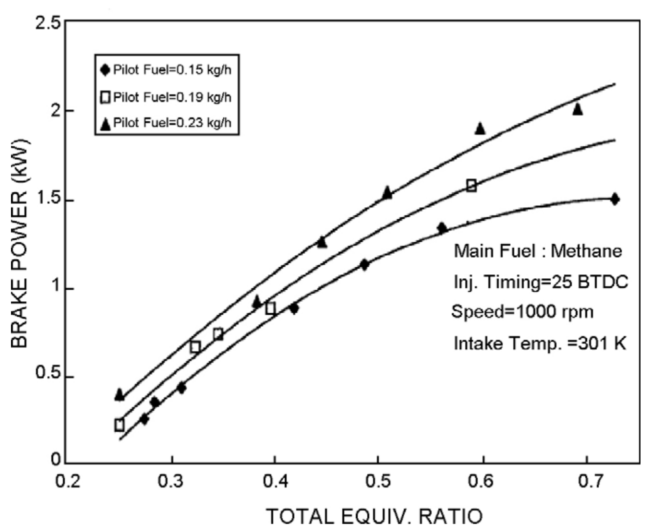

Fig. 55. Variations of experimental results of brake power with total equivalence ratio for different values of pilot fuel quantities (main fuel: methane) [36].

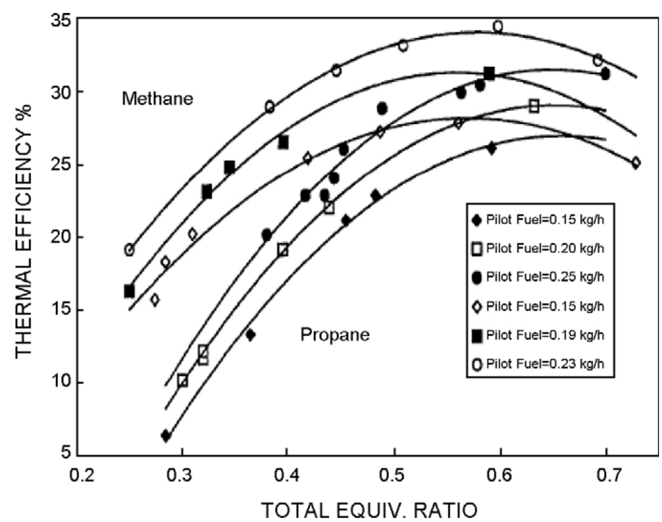

Fig. 58. Variations of thermal efficiency with total equivalence ratio with different values of pilot fuel quantities [36].

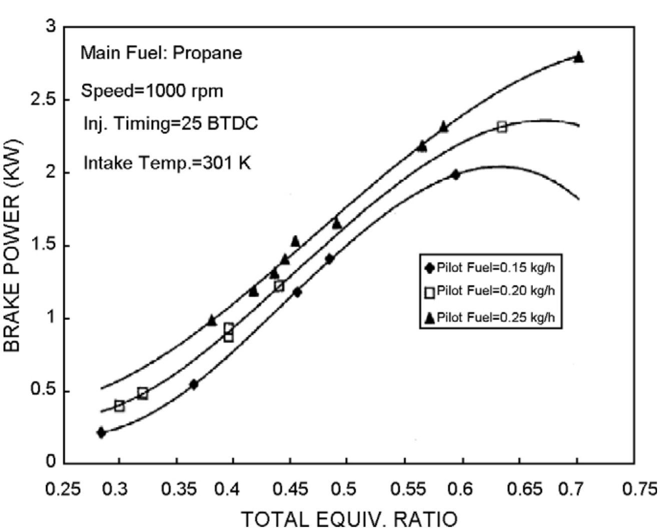

Fig. 56. Variations of experimental results of brake power with total equivalence ratio for different values of pilot fuel quantities (main fuel: propane) [36].

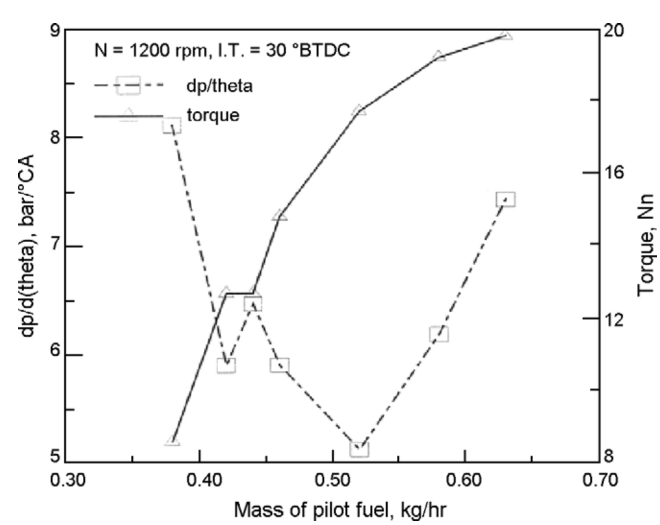

Fig. 59. Effect of pilot fuel mass on pressure rise rate for the dual-fuel engine [14].

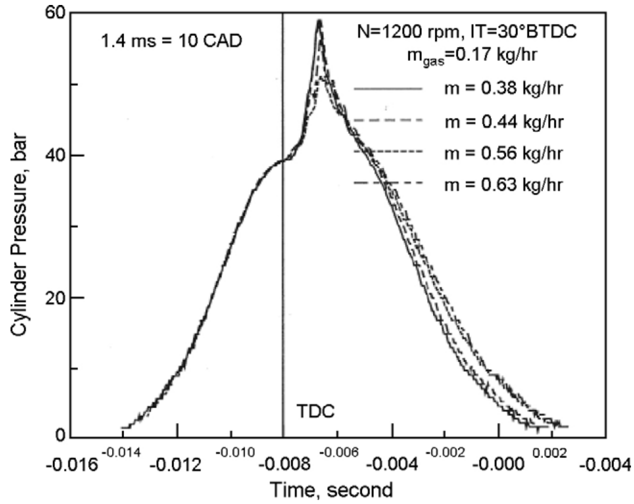

Fig. 60. Pressure–crank diagram for dual-fuel engine, at different pilot fuel mass [14].

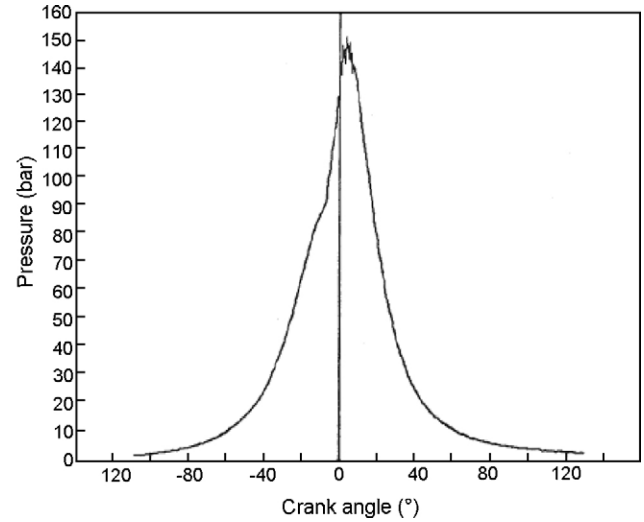

Fig. 62. Pressure–crank angle diagram of diesel fuel operation [37].

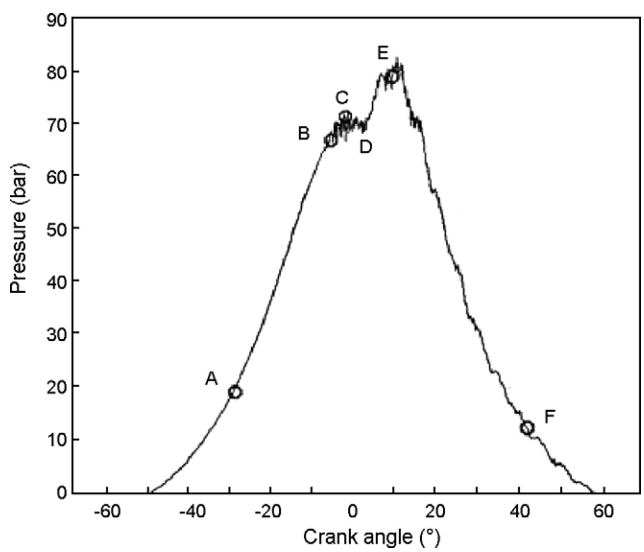

Fig. 61. Dual-fuel pilot injection pressure–crank angle diagram [37].

injection systems experience poor atomization and combustion when the amount of fuel injected per cycle is reduced below 5–10% of the maximum design level.

The variations of the exhaust gas concentrations of methane and carbon monoxide with total equivalence ratio for different pilot fuel quantities is investigated by Badr et al. [9]. Fig. 47(a) and (b) shows that there is a limiting equivalence ratio beyond which the exhaust emissions of the carbon monoxide and the unconverted methane become virtually unaffected by the pilot quantity. This is indicative of the equivalence ratio limit for successful flame propagation from the pilot ignition centres. The variations of the flame spread limits (FSL) with changes in pilot quantity are shown in Fig. 48. The increase in pilot quantity lowers the FSL due to number of contributing factors. These include; greater energy release on ignition, correspondingly improved pilot fuel injection characteristics, larger pilot-mixture envelope size, larger ignition centres, higher rate of heat transfer to the unburned gaseous mixture and increased contribution of hot residual gases. The flame initiation limit exhibits a similar trend is also shown in Fig. 48.

Abd et al. [36] have investigated the effect of three pilot fuel quantities, 0.15, 0.20 and 0.25 kg/h, on the performance and emissions of an indirect injection diesel engine fuelled with gaseous fuel. Fig. 49 shows that at very light loads with a small pilot fuel quantity, the concentrations of UBHCs measured are relatively high. This is because, in an excessively lean mixture, the flame originating from the pilot ignition (if there is any) cannot propagate throughout the whole combustion chamber; only partial oxidation occurs, and thus UBHC emissions as well as CO emissions are relatively higher, as shown in Fig. 50. At higher loads, when the gaseous fuel concentration in the air charge is above the lean combustion limit, the flame is able to propagate through most of the combustion chamber unaided, and varying the pilot fuel quantity has little effect. The change in oxidation reactions from unsuccessful to successful flame propagation reduces the HCs and CO emissions slightly as shown in Figs. 51 and 52.

In dual-fuel engines, the effective size of the combustion zone, which relates to the size of the pilot fuel zone, is another important

Table 6
Effect of pilot fuel/gas ratio on knock characteristics of dual-fuel engine.

Load (N)	Pilot fuel (kg/h)	Gas supply (kg/h)	Total fuel (kg/h)	% pilot fuel	% gas supply	Mixture strength
6.61	0.305	0.626	0.931	32.76	67.24	0.557
13.18	0.292	0.720	1.012	28.85	71.15	0.620
17.72	0.271	0.792	1.063	25.49	74.51	0.667
23.78	0.295	0.846	1.141	25.85	74.15	0.730
28.82	0.313	0.882	1.195	26.19	73.81	0.779
33.87	0.331	0.907	1.238	26.74	73.26	0.823
38.92	0.343	0.936	1.279	26.82	73.18	0.866
44.47	0.307	1.008	1.315	23.35	76.65	0.922

Engine speed = 3000 rev/min [37].

factor, besides to the mechanism of oxidation of nitrogen that determines the quantity of nitrogen oxides produced. For the same total equivalence ratio, increasing the pilot fuel quantity increases the charge temperature which tends to increase the production of NO_x, as shown in Fig. 53. For relatively high loads, the combustion of gaseous fuel is more complete and less affected by the pilot fuel quantity, and thus, has a mild effect. Similarly, for a given pilot fuel quantity when higher gaseous fuel concentrations in the cylinder charge are employed, the significant increases in the size of the combustion zone lead to correspondingly increased higher production of NO_x. Fig. 54 shows that the production of NO_x is influenced markedly by both the quantity of the pilot fuel employed and the overall equivalence ratio. The use of large pilot fuel quantities and high charge equivalence ratios results in a significant increase in the production of NO_x. Figs. 55 and 56 indicate that the employment of a large pilot fuel quantity produce higher power output. The increase of pilot fuel quantity leads to successful flame propagation and, consequently, increases the output power. Fig. 57 indicates that using a greater pilot fuel quantity, to enhance the combustion process at low loads, leads to increase the tendency to knock at high loads. The thermal efficiency for methane is seen to be greater than that for propane at low loads (Fig. 58). This is due to the higher ignition delay of propane at low loads.

The experimental investigation conducted by Selim [14] shows that increasing the pilot diesel fuel mass can resulted in increase in the engine torque (Fig. 59). This is postulated to the increase in the heat released from burning more fuel. The combustion noise ($dP/d\theta$), however, dropped with the increase of pilot mass from around 8.1 bar/°C A at 0.38 kg/h pilot mass to about 5.12 bar/°C A at 0.52 kg/h. Pressure rise rate then increases to 7.42 bar/°C A as the pilot fuel mass increases to 0.63 kg/h. This in turn, increases in flame volume resulting smooth burn of methane gas smoothly and at a lower rate of combustion. The maximum pressure (Fig. 60) and the maximum rate of pressure rise ($dP/d\theta$) are lower for a pilot fuel mass of 0.52 kg/h, and it is increased for a lower or higher amount of diesel pilot fuel.

The effect of pilot fuel/gas ratio on knock characteristics of dual-fuel engine is examined by Nwafor [37]. The knock characteristics for pure diesel and dual fuel operations are compared in Figs. 61 and 62. Similar figures are also presented by the author for natural gas in and unmodified CI engine [38]. The ripples in Fig. 61 are the indication of combustion knock. Dual-fuel operation shows longer ignition delay as measured by the author. The degree of knock in this phase depends on the ratio of the alternative fuel (NG) to the pilot fuel as given in Table 6 and thus on the load and speed of operation. The increase in speed increases the ignition delay when running on pure diesel fuel, hence the quantity of premixed pilot fuel that takes part in combustion increases. Increasing the pilot fuel and reducing primary fuel reduces the knocking phenomena in dual-fuel engines.

Selim [4] has examined the effects of pilot fuel quantity of a dual fuel engine are shown in Fig. 63(a)–(d). During these experiments the following parameters are kept constant: engine speed 1300 rpm, pilot fuel injection timing 35° BTDC and compression ratio 22. Increasing the quantity of pilot diesel fuel increases the torque output, Fig. 63(a) and (b), and, hence, thermal efficiency for the three gaseous fuels used. Increasing the pilot diesel fuel for the

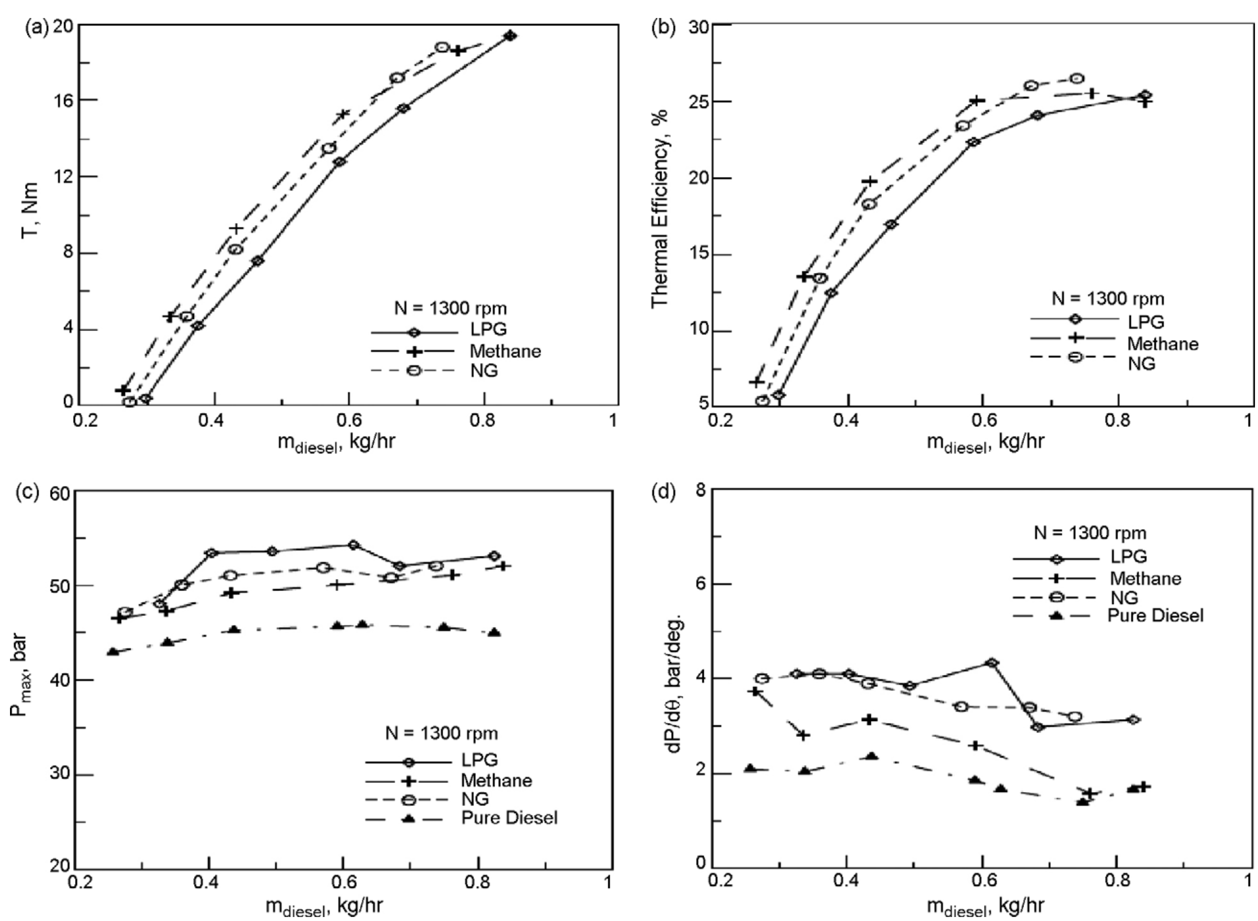

Fig. 63. Effects of pilot fuel mass on performance and noise: N = 1300 rpm, IT = 35° BTDC, CR = 22 [4].

three cases results in greater energy release on ignition, improved pilot injection characteristics, larger size of pilot mixture envelope with greater entrainment of the gaseous fuel, a larger number of ignition centres requiring shorter flame travels and a higher rate of heat transfer to the unburned gaseous fuel–air mixture [9]. These factors tend to increase the power output and thermal efficiency of the dual-fuel engine [36]. Increasing the pilot fuel mass also resulted in higher maximum combustion pressure as may be seen in Fig. 63c. For the dual-fuel engine, the maximum pressure is always higher than in the diesel fuel case due to the combustion and extra heat released from gaseous fuels. The maximum pressure rise rate, as seen in Fig. 63d, is generally reduced when the pilot fuel quantity is increased. The decrease in the combustion noise ($dP/d\theta$) when the pilot fuel mass is first increased is postulated to be due to the increase in flame volume resulting from the increase in pilot fuel mass, which burns the gaseous fuel smoothly and at a lower rate of combustion. However, when the pilot fuel mass increases beyond a certain amount, the ignition delay period of the pilot diesel increases and hence, the pressure rise rate ($dP/d\theta$) for the gas–air mixture increases [14].

5.5. Effect of engine compression ratio

Selim [4] has examined the effects of compression ratio of a dual-fuel engine: First, on knock onset and ignition failure (Fig. 64a–c) and secondly, on the maximum pressure rise rate (Fig. 65a–c). For LPG, reduction in the compression ratio results in retarding the occurrence of knock onset in the dual-fuel engine (from 8.1 to 17.6 to 20 N-m) and also extended the ignition limits greatly (from 7.85 to 17 to 18.5 N-m) as shown in Fig. 64a–c. This is due to the early knocking at high compression ratios associated with higher pressures and temperatures and lower self-ignition temperatures of LPG. For extended ignition limits and knock free operation of the dual-fuel engine, the compression ratio has to be reduced to lower values. For the NG mixture and methane, Fig. 64(b)–(c), have similar trend to LPG with the only difference at the compression ratio of 22. As LPG has the lowest self-ignition temperature (about 400 °C), it starts knocking and ignition fails at lower engine torque compared to the other two gases namely, methane (about 650 °C) and CNG (about 500 °C).

The maximum pressure rise rate at different compression ratios for the three fuels is illustrated in Fig. 65(a–c). It is seen that increasing the compression ratio generally increases the combustion noise due to the higher self-ignition possibility of the gaseous fuels at higher pressures and temperatures. As the compression ratio is reduced, the combustion noise is also reduced, and the ignition limits are extended.

5.6. Effect of intake manifold conditions

In order to improve exhaust emissions at part load, Kusaka et al. [39] have modified the intake charge condition including intake temperature and exhaust gas recirculation (EGR) as given in Table 7. They have installed a heat exchanger in the intake system and a Pt-catalyst is used in the exhaust system to reduce unburned natural gas emission.

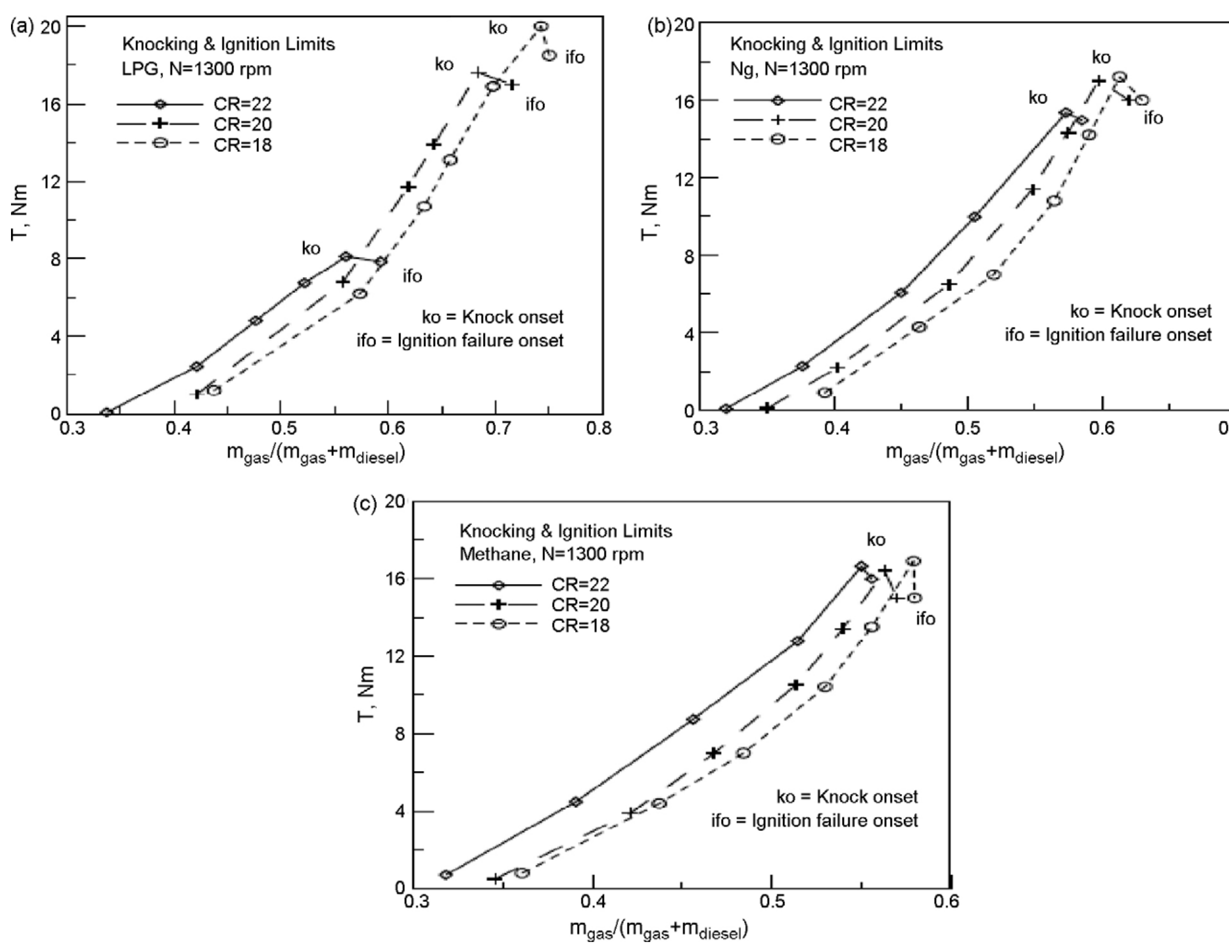

Fig. 64. Effect of compression ratio on knock and ignition limits for LPG, CH_4 and NG: N = 1300 rpm, IT = 35° BTDC, m_d = 0.37 kg/h [4].

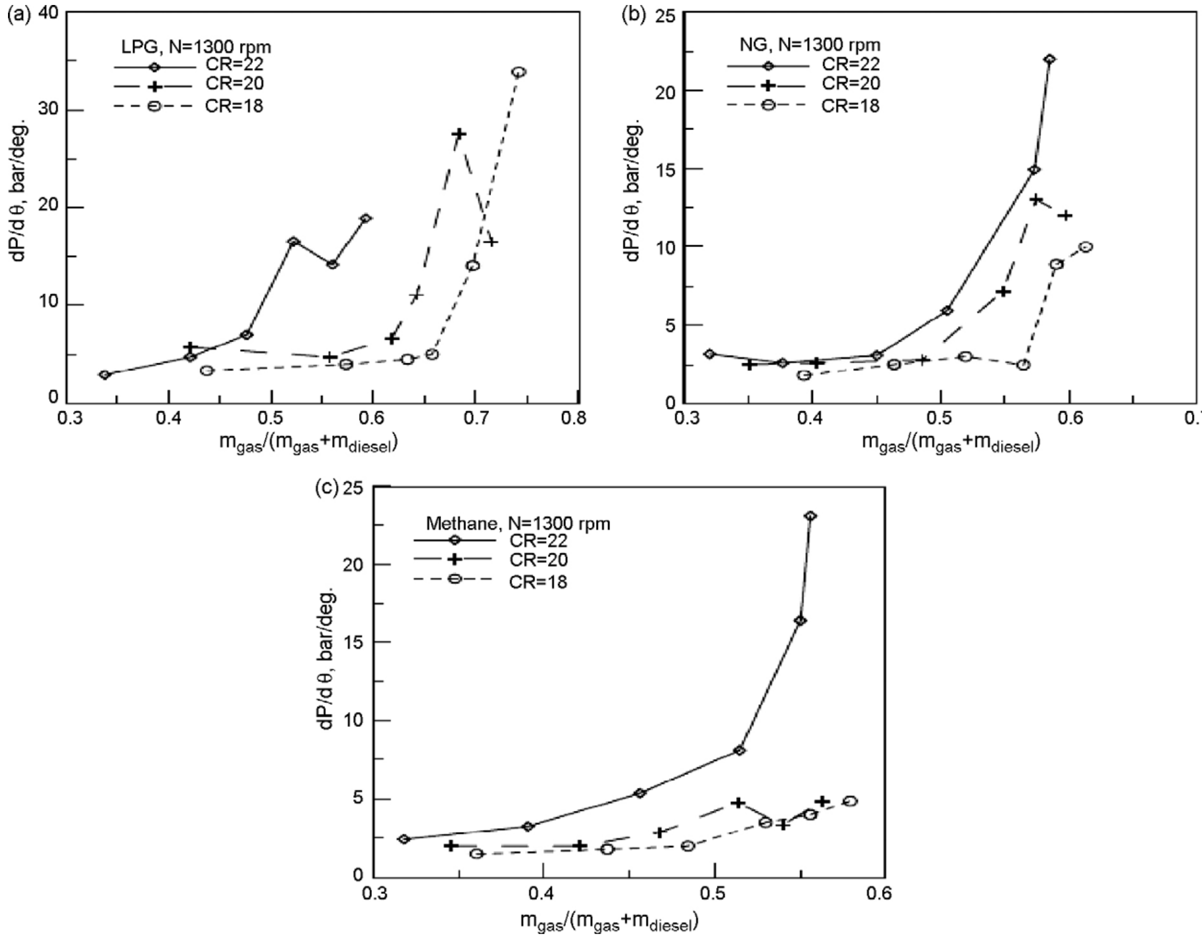

Fig. 65. Effect of compression ratio on combustion noise for LPG, CH_4 and NG: N = 1300 rpm, IT = 35° BTDC, m_d = 0.37 kg/h [4].

The base line condition is taken here as; intake temperature of 20 °C and EGR 0%. In the case of intake heating of 165 °C, the start of RHR becomes earlier than that of other conditions due to higher cylinder temperature (Fig. 66). On the other hand, the combination of 50% EGR and intake heating of 210 °C at the start of RHR becomes retarded compared to that of intake heating of 165 °C without EGR. Ignition reaction proceeds faster by intake heating, while the inert gases included in re-circulated exhaust gas impede progress of ignition reactions. As a result of the competition of these above two conflicting effects, the ignition timing is determined as shown in Fig. 66. It is also seen from this figure that EGR can favourably control the rate of pressure rise. The reason why the shape of RHR at intake heating of 165 and 210 °C has two peaks is that the natural gas combustion takes place after the combustion of pilot diesel is activated, due to the increased natural gas mixture temperature.

Table 7
Engine tests conditions for intake heating combined with EGR (engine speed: 1280 rpm, load: 1/5, NG fraction: 80%) [39].

EGR ratio (%)	Diesel injection timing (degree BTDC)	Intake temperature (°C)	
		Before the heat exchanger	After the heat exchanger
30	13	60	170
50	13	110	210
60	13	145	230

Fig. 67 shows the exhaust emissions characteristics under various conditions. In this figure, "Diesel" means diesel operation without dual-fuelling, while "Base" represents baseline condition. As shown in this figure, EGR with intake heating can favourably reduce THC emissions compared to the baseline and therefore, thermal efficiency is improved. NO_x is drastically increased in the case of intake heating without EGR. However, when EGR is combined with intake heating, NO_x is reduced drastically. In the dual-fuel system, the engine can operate with high EGR ratio because dual-fuel diesel engine produces little soot, leading to reduced NO_x emission. This is because of the fact that the inert gas, which has a large heat capacity, lowers combustion temperature, while EGR reduces the oxygen concentration in the cylinder. From these above points of view, EGR combined with intake heating reduces NO_x and THC emissions without deteriorating engine thermal efficiency.

5.7. Effect of type of gaseous fuel

Laboratory investigation is carried out by Bari [40] to see the effect of carbon dioxide (CO_2) on the performance of biogas-diesel duel-fuel engine. Mixing of NG with CO_2 at different level of compositions is the main part of the experiment. At certain percentage of diesel substitution by NG (15, 30, 50 and 75%), the pure CO_2 is introduced with the NG. The speed and power are maintained at constant level by varying the quantity of diesel flow. When the percentage of CO_2 in biogas increases, the biogas supply to produce the same power also increases (Fig. 68). The rate of

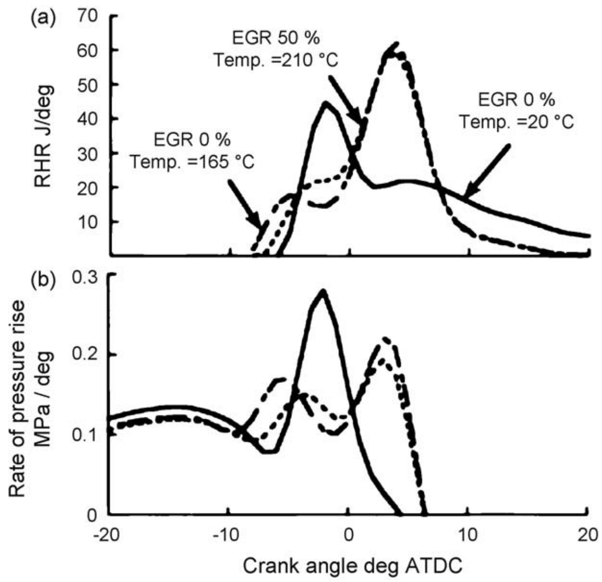

Fig. 66. The effect of intake heating and EGR on (a) R.H.R and (b) rate of pressure rise (engine speed: 1280 rpm, load: 1/5, NG fraction: 80%) [39].

increase of biogas at higher substitution is higher. Figs. 69 and 70 show the variation of diesel flow and 'bsfc' with variation of CO_2 in the biogas. The CO_2 in the mixture dissociate into CO and oxygen, because the flame temperature of diesel is very high to initiate dissociation. It is evident that CO is comparatively fast burning gas than other alternative fuels [41]. Thus, the burning rate of the total gas–air mixture accelerates for the presence of CO. The amount of oxygen found by dissociation of CO_2 increases the concentration of oxygen in the gas air mixture. Therefore, it reduces the ignition delay as well as enhances the combustion of unburned carbon particles. Under the above conditions, the engine performance is found comparatively better with lower 'bsfc's and diesel flow rates. Therefore, the trend of the 'bsfc's curve is falling up to 20–30% CO_2 in the gaseous mixture. When the percentage of CO_2 becomes higher in the gaseous mixture, CO_2 remains undissociated. This undissociated CO_2 acts as inert gas. Addition of such inert gas affects the burning velocity of gas–air mixture [42]. Therefore, it results incomplete combustion and increases 'bsfc'. Biogas containing more than 40% CO_2 needs scrubbing, because it is found in this research that the engine ran harshly with biogas containing high CO_2 (>40%).

Henham and Makkar [3] have conducted tests on the dual-fuel diesel engine at various proportions of gas mixture comprising of NG and CO_2. First, gasoil is substituted by NG from the British mains (94% methane, 34.8 MJ/m^3) at four constant levels (22%, 37%, 45% and 58%). Then, taking each constant level of NG as 100%, it is mixed with CO_2 to vary the composition of gas mixture. Test results in Figs. 71–73 are at engine speed 2000 rev/min and torque 40 N-m using NG:CO_2 mixture for a range of 100:0–40:60 at four constant NG substitution levels. The overall efficiency falls with NG substitution at all constant levels (Fig. 71). On mixing NG with CO_2 efficiency is not much affected up to 37% NG substitution. With higher NG substitution, efficiency decreases with increasing CO_2 in gas mixture. At 58% NG substitution level, efficiency decreases from 28.2% to 26.2% with increasing CO_2 in gas mixture. With higher gas substitution a greater proportion of air is replaced by gas. So, volumetric efficiency of the engine is lowered and it results in less power. Fig. 72 indicates that exhaust temperature is affected more by NG substitution up to 45%. At 58% NG substitution, exhaust temperature increases with increasing CO_2 in gas mixture, from 382 to 402 °C. The CO is affected mainly by NG substitution and not so much by the proportion of CO_2 in the gas mixture (Fig. 73). The increase in CO as compared to that with gasoil only (when it is only 0.04%) is caused by lower effective air–fuel ratio as gas mixture replaces more air. Test results in Figs. 74–76 are at engine speed 2800 rev/min and torque 40 N-m using NG:CO_2

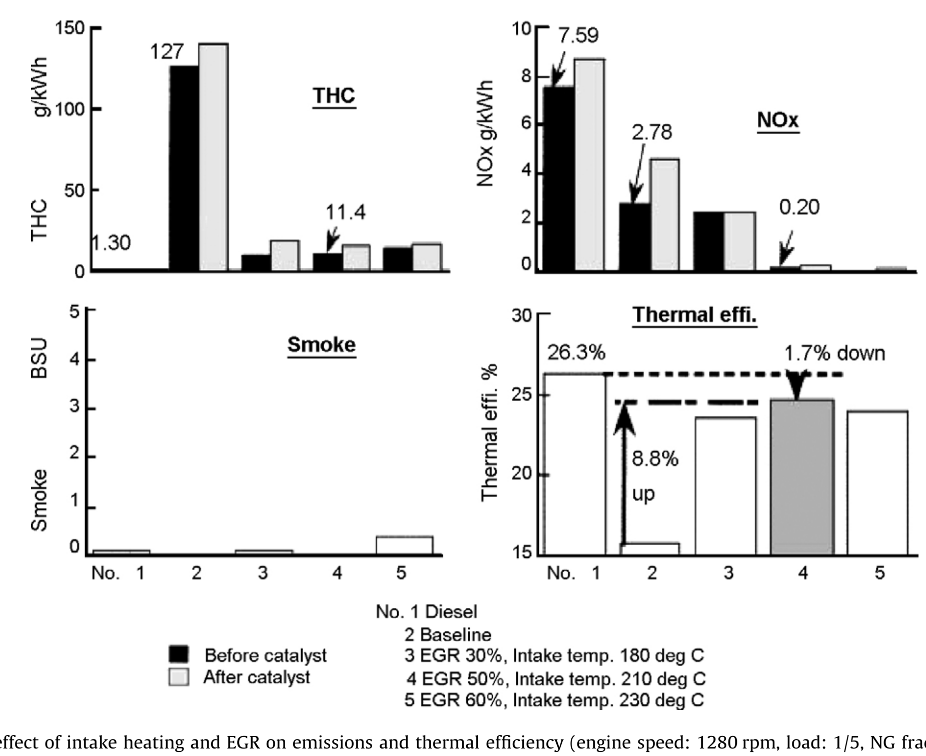

Fig. 67. The effect of intake heating and EGR on emissions and thermal efficiency (engine speed: 1280 rpm, load: 1/5, NG fraction: 80%) [39].

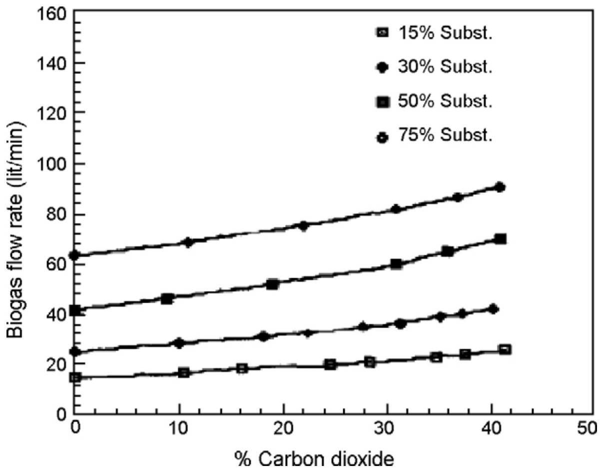

Fig. 68. Variation of biogas flow with the increase of CO_2 in biogas [40].

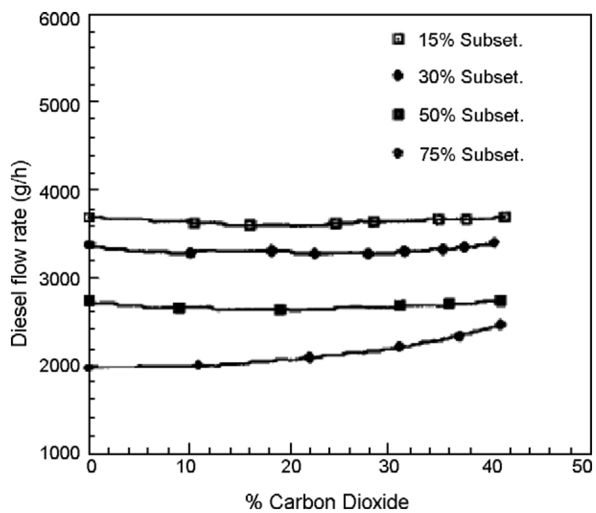

Fig. 69. Variation of diesel flow with the increase of CO_2 in biogas [40].

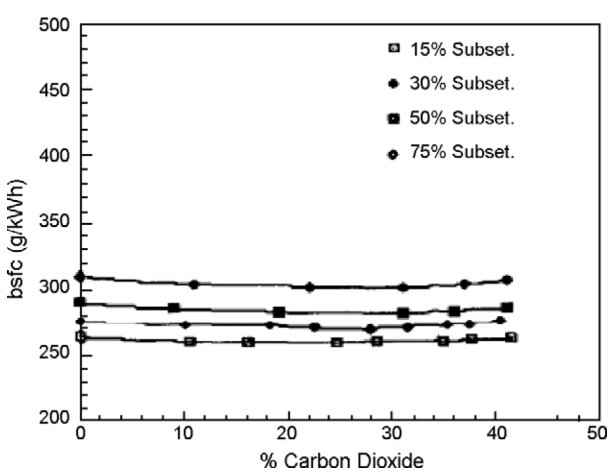

Fig. 70. Variation of 'bsfc' with the increase of CO_2 in biogas [40].

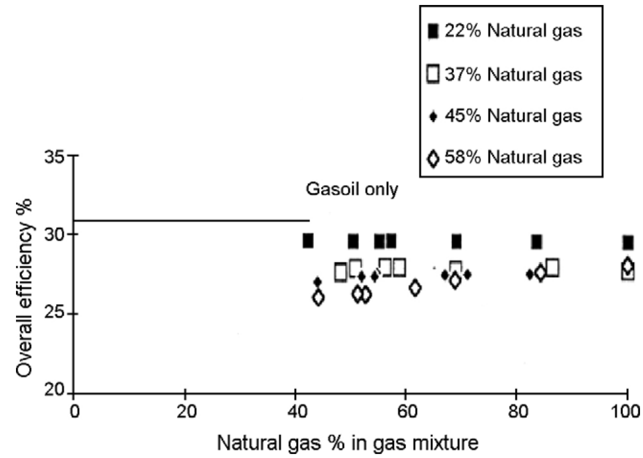

Fig. 71. IDI dual-fuel efficiency variation using gas mixture at 2000 rev/min and 40 N-m [3].

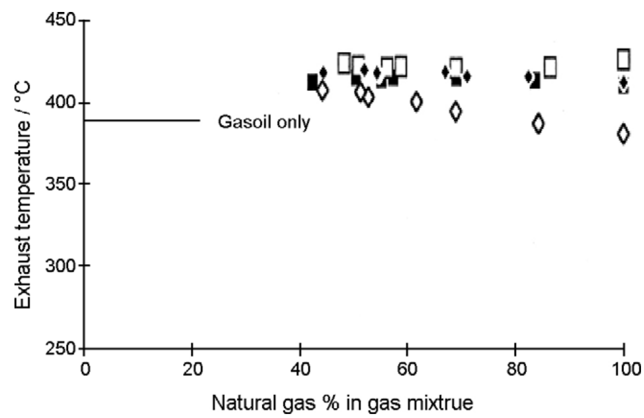

Fig. 72. IDI dual-fuel exhaust temperature variation using gas mixture at 2000 rev/min and 40 N-m [3].

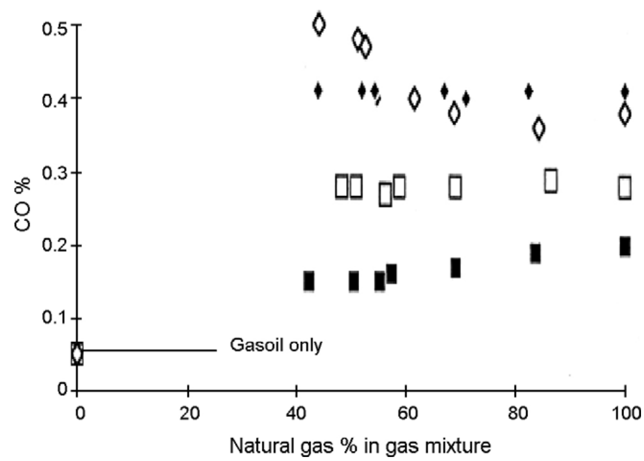

Fig. 73. IDI dual-fuel CO variation using gas mixture at 2000 rev/min and 40 N-m [3].

mixture for a range of 100:0–30:70 at five constant NG substitution levels. Fig. 74 indicates that the overall efficiency decreases with increase in CO_2 in gas mixture at all substitution level where as the exhaust temperature and CO follow the same patterns as at 2000 rev/min except at 65% NG substitution (Figs. 75 and 76). In this condition, the combustion is less controlled and knock is noticed during the test run. The in-cylinder pressure characteristics of the test engine for gasoil only, gasoil and 58% NG substitution, and gasoil and gas mixture ($NG:CO_2$, 1:1), respectively are presented in Figs. 77–82. At 2000 rev/min, peak pressure

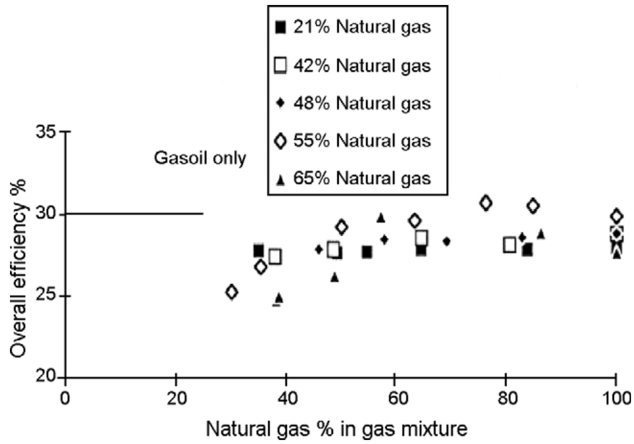

Fig. 74. IDI dual-fuel efficiency variation using gas mixture at 2800 rev/min and 40 N-m [3].

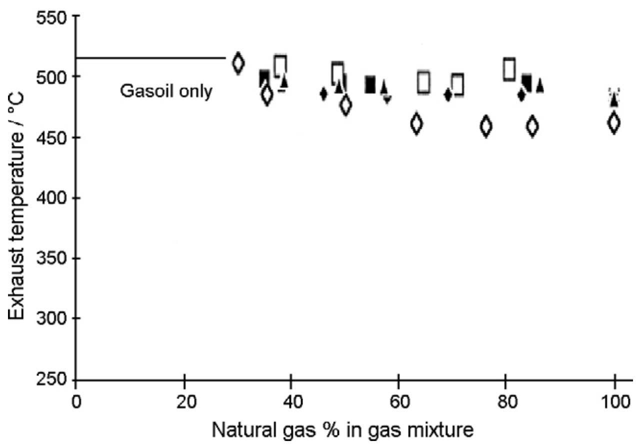

Fig. 75. IDI dual-fuel exhaust temperature variation using gas mixture at 2800 rpm and 40 N-m [3].

rises from 70 bar to 83 bar at 60% NG substitution and falls to 77 bar for gas mixture of NG:CO_2 (1:1). Sharper peaks are observed in Figs. 78 and 79 compared to Fig. 77 are due to be the result more fuel availability at the initiation of combustion. For 2800 rev/min, peak pressure rises from 57 bar to 70 bar at 60% NG substitution and falls to 67 bar for gas mixture of NG:CO_2 (1:1) indicated in Figs. 80–82.

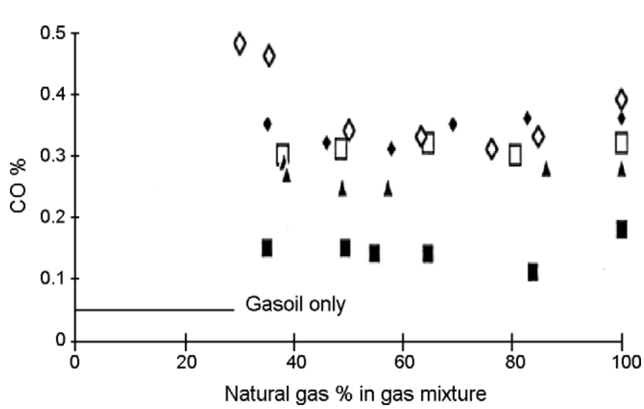

Fig. 76. IDI dual-fuel CO variation using gas mixture at 2800 rev/min and 40 N-m [3].

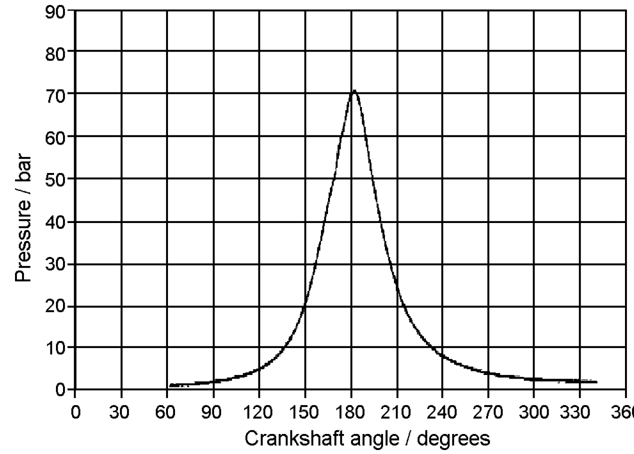

Fig. 77. *P–θ* diagram for gasoil only @ 2000 rev/min, 40 N-m [3].

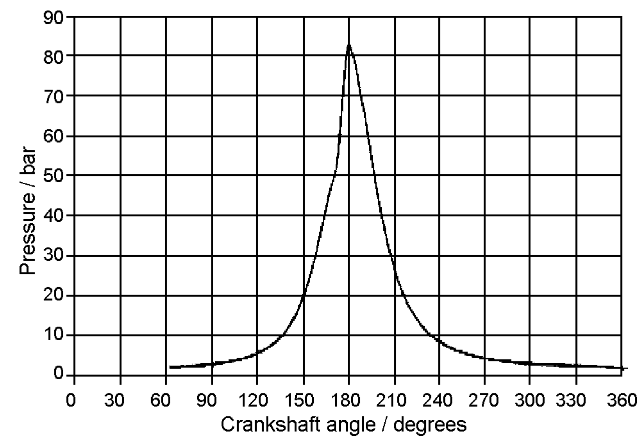

Fig. 78. *P–θ* diagram for gasoil and 60% NG substitution @ 2000 rev/min, 40 N-m [3].

Papagiannakis and Hountalas [24] have examined the dual-fuel engine performance and exhaust emission characteristics by varying the mass ratio of gaseous fuel (NG). The engine is supplied with NG from the local low-pressure distribution network after making the appropriate modifications. Measurements are taken at three different engine loads corresponding to 40%, 60% and 80% of full load and three engine speeds of 1500,

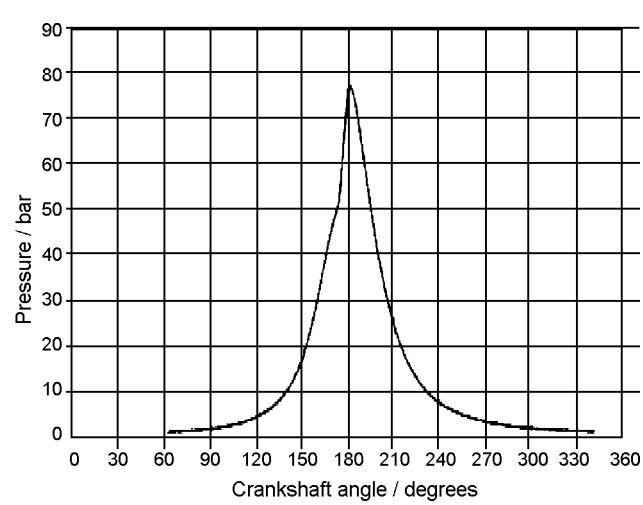

Fig. 79. *P–θ* diagram for gasoil and NG:CO_2 (1:1) @ 2000 rev/min, 40 N-m [3].

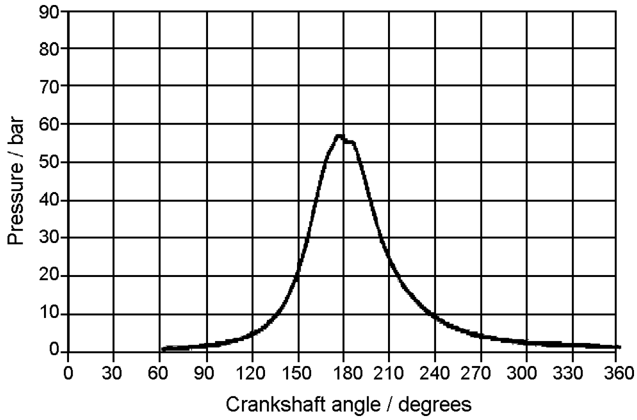

Fig. 80. P–θ diagram for gasoil only @ 2800 rev/min, 40 N-m [3].

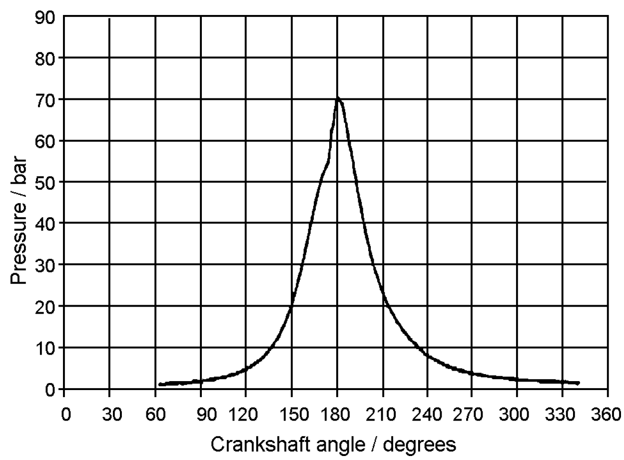

Fig. 81. P–θ diagram for gasoil and 60% NG substitution @ 2800 rev/min, 40 N-m [3].

2000 and 2500 rpm under both, normal diesel operation (100% diesel fuel) and dual-fuel operation (NG and diesel fuel). At part load condition, increasing the NG mass ratio over 50%, the cylinder pressure after ignition increases at a slower rate compared to normal diesel operation (Fig. 83). The initial heat release rate for mass ratios up to 50% is almost the same while for higher values it is lower, revealing less premixed combustion. The effect of NG combustion on the total heat release becomes evident only at high mass ratios around 80%. At high engine load (80%) as shown in Fig. 84, the cylinder pressure traces under

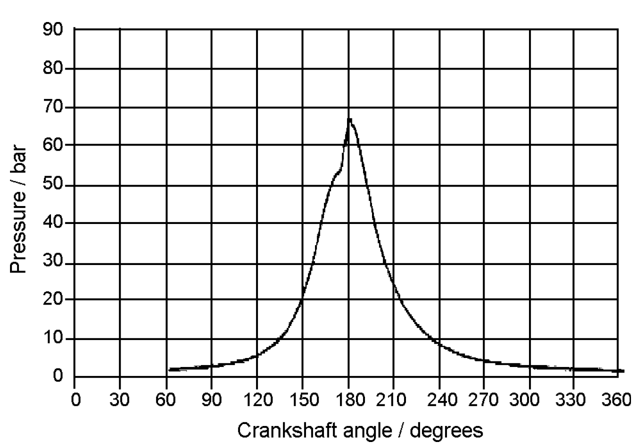

Fig. 82. P–θ diagram for gasoil and NG:CO_2 (1:1) @ 2800 rev/min, 40 N-m [3].

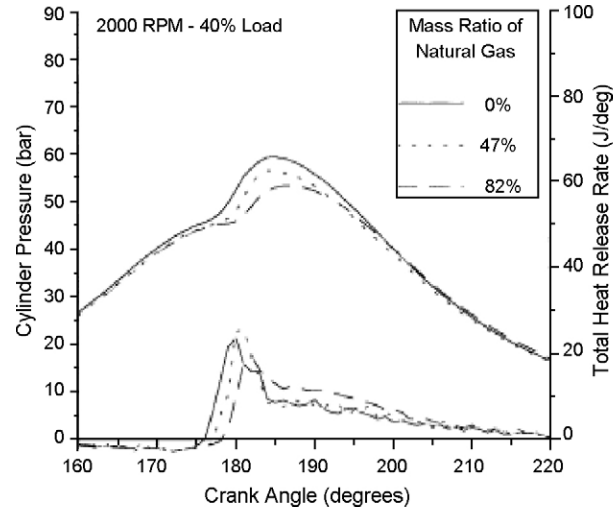

Fig. 83. Experimental cylinder pressure and total heat release traces under normal diesel and dual-fuel operation for 2000 rpm engine speed at 40% of engine load [24].

dual-fuel operation strongly diverge from the respective values under normal diesel operation as NG mass ratio increases. The second peak of heat release observed in this figure is due to combustion of NG. But, it does not affect strongly the pressure curve since it is in the expansion stroke and causes only a slight increase of its value. For all cases examined the cylinder pressure under dual-fuel operation decreases when increasing the amount of NG. This is observed during compression and the initial stages of combustion and is mainly the result of the higher specific heat capacity of the NG–air mixture and its slower combustion rate compared to that of diesel fuel.

Referring to Fig. 85, maximum combustion pressure is strongly affected by the presence of gaseous fuel. As the quantity of gaseous fuel increases, keeping engine load constant, peak cylinder pressure decreases significantly while the slope remains almost the same regardless of engine load. Ignition delay increases with increase in the percentage mass of NG (Fig. 86). The increase is due to the reduction of charge temperature close to the point of fuel injection and high overall specific heat capacity [11,27]. Combustion duration under dual-fuel operation is

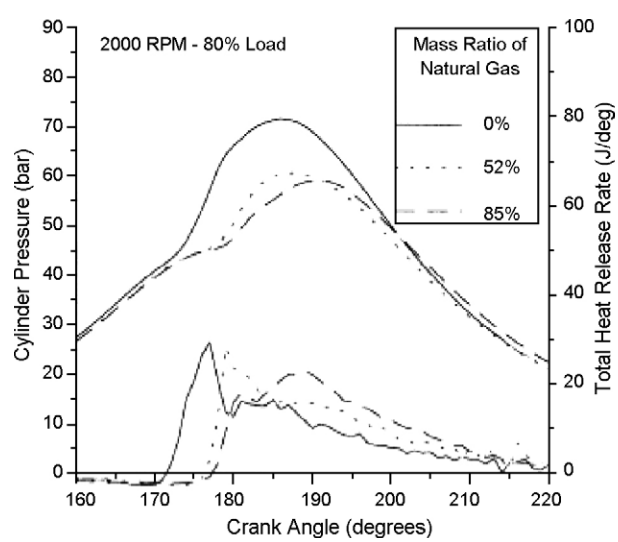

Fig. 84. Experimental cylinder pressure and total heat release traces under normal diesel and dual-fuel operation for 2000 rpm engine speed at 80% of engine load [24].

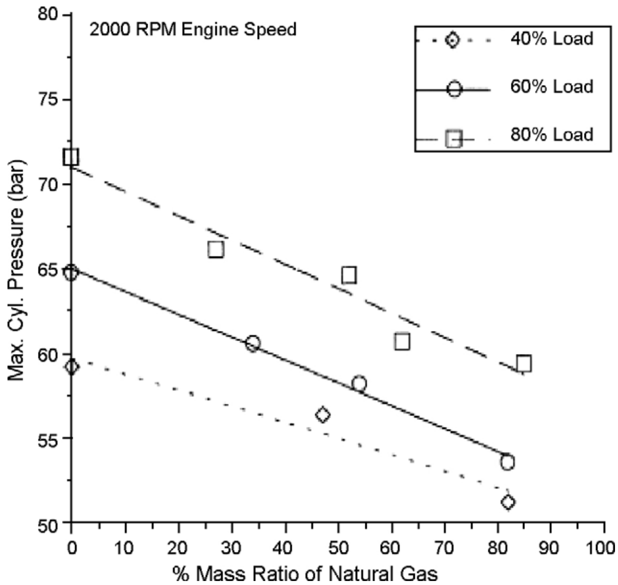

Fig. 85. Maximum combustion pressure as function of NG mass ratio at 2000 rpm for various engine loads [24].

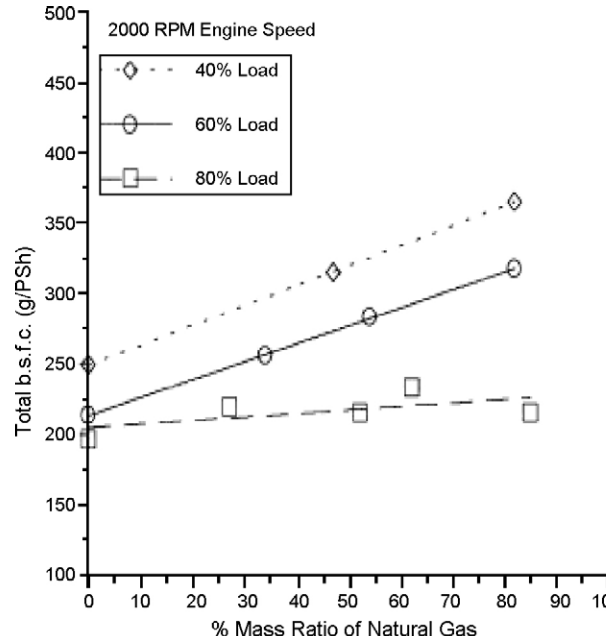

Fig. 87. Variation of 'bsfc' as function of NG mass ratio at 2000 rpm for various engine loads [24].

generally longer compared to the one under normal diesel operation for all cases examined (Fig. 86). Under dual-fuel operation, there is an increase of combustion duration with increasing NG mass ratio at a rate of which is slightly more intense at low mass ratios of NG. At part load under dual-fuel operation, the total 'bsfc' is considerably higher compared to the one under normal diesel operation, mainly as a result of the low combustion rate of gaseous fuel (Fig. 87). A similar trend is observed at high load under dual-fuel operation, but in this case the slope of 'bsfc' increment with NG mass ratio is lower compared to the one at part load. Here, the total 'bsfc' is estimated from the brake power output of the engine and the measured mass flow rate of fuels. Thus, no correction is made to consider for the difference in lower heating values (LHV) of NG and diesel fuel. Considering the fact that the LHV of NG is higher compared to the one of diesel fuel used, its total 'bsfc' is even higher if these are corrected to the LHV of the diesel fuel. This reveals a poor utilization of the gaseous fuel due mainly to the lower temperature inside the combustion chamber of the engine and the late start of ignition because of the higher ignition delay [43].

At part load as the NG mass ratio increases, soot concentration decreases sharply since less liquid fuel is injected on a percentage basis and thus less soot is formed (Fig. 88). With high engine load and low gaseous fuel mass ratios, the charge temperature is

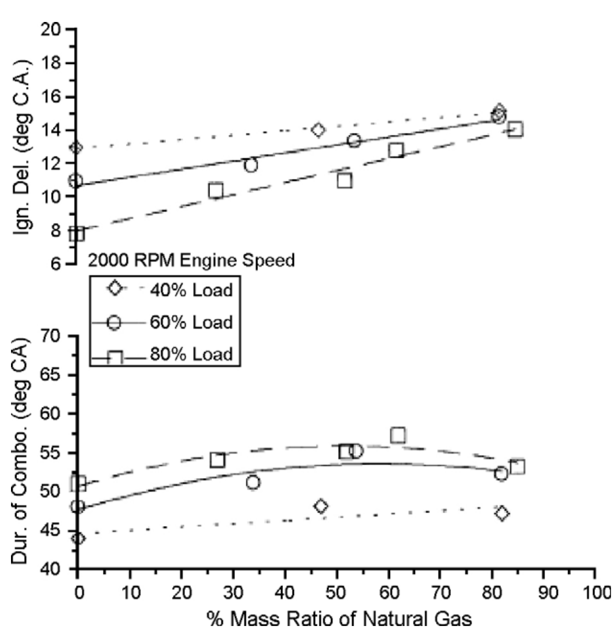

Fig. 86. Variation of ignition delay and combustion duration as function of NG mass ratio at 2000 rpm for various engine loads [24].

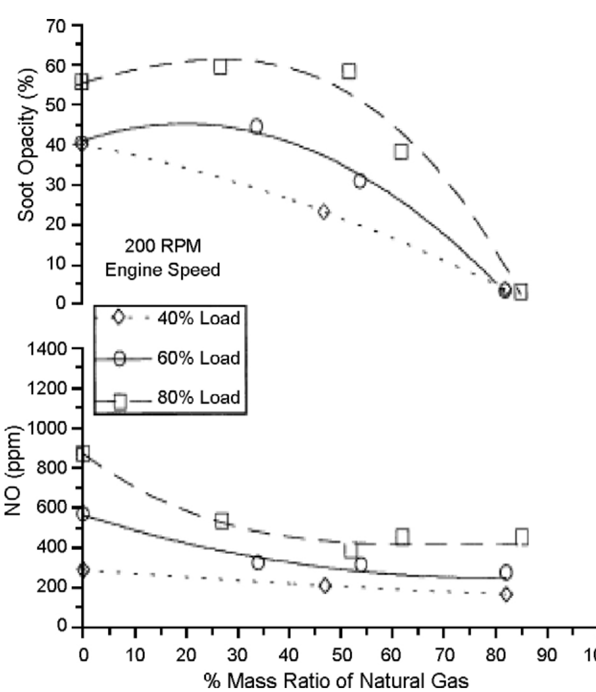

Fig. 88. Soot opacity and nitric oxide as function of NG mass ratio at 2000 rpm for various engine loads [20].

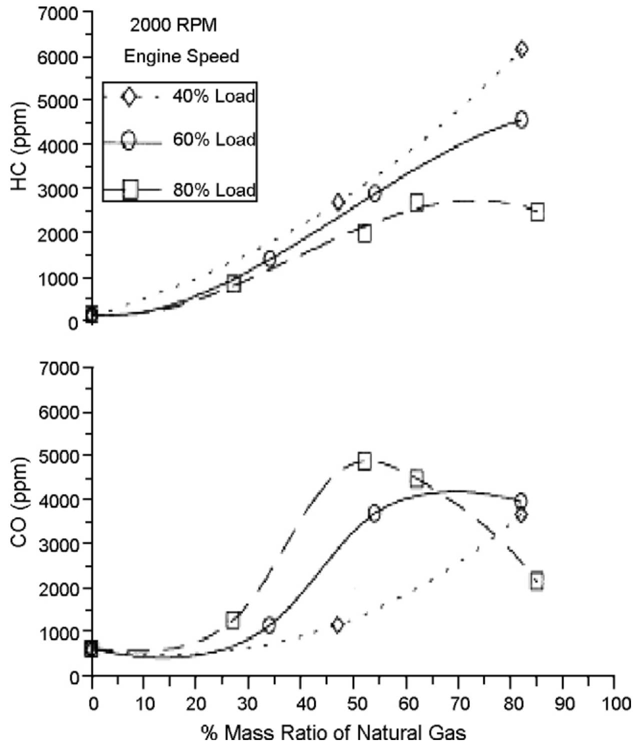

Fig. 89. UBHC and CO as function of NG mass ratio at 2000 rpm for various engine loads [24].

inferior and it results a slight increase in soot emissions compared to the one under normal diesel operation. But, at higher mass ratios soot decreases and this reduction compared to normal diesel operation is sharper. This is due to the higher gas temperature that promotes soot oxidation rate contributing to a further decrease of soot concentration. Taking these into consideration for dual-fuel operation, use of a high percentage of NG is an efficient way to reduce soot concentration. Practically, the gaseous fuel produces no soot while it contributes to the oxidation of the one formed from the combustion of the liquid fuel. It is widely recognised that the high oxygen concentrations and high charge temperatures favour the formation of nitric oxide (NO) [22,23,44,45]. The increase of NG mass ratio results to a decrease of NO concentration compared to the one under normal diesel operation (Fig. 88). At high engine load and low mass ratios of NG there is a sharper decrease of NO concentration, compared to part load conditions with increasing gaseous fuel percentage. The reduction of NO concentration is due to several factors; the less intense premixed combustion, the reduction of gas temperature due to increase of the specific heat capacity, the slower combustion and finally, the reduction of oxygen concentration due to presence of the NG mass ratio which replaces an equal amount of air in the cylinder charge. The variation of UBHC in the exhaust gases is consistent with the quality of the combustion process of the engine [11,25,27,45]. At low load, HC emissions increase with increase of gaseous fuel percentage (Fig. 89). The low charge temperature combined with a slower burning rate leads to an increase of gaseous fuel, which does not burn completely. At high engine load there is an increase of HC

Table 8
Comparison of results by researcher(s) on the performance of dual-fuel diesel engines.

Researcher(s)	Performance of dual-fuel diesel engine
Effect of engine load	
Selim [14]	Combustion noise increases with increase in load and is always higher than that of for the diesel fuel case
Uma et al. [16]	The engine performance decreases at part load conditions both at diesel and dual-fuel mode. CO emissions are higher than diesel engines at all operated load conditions. HC emissions are little higher than diesel case. But, dual-fuel operation reduces NO_x and SO_2 without increasing particulate emissions
Papagiannakis and Hountalas [19]	Total BSFC is inferior compared to the normal diesel operation at the same engine operating conditions. Lower peak cylinder pressure compared to the normal diesel operation at a given load condition. At low load, the combustion duration is longer compared to normal diesel operation, while at high load, it is shorter. Lower NO_x, drastic decrease in soot emissions, but considerably higher CO and HC emissions
Singh et al. [30]	A minor reduction in engine output (1–2%) compared to the diesel case. The CO, CO_2, NO and NO_2 reduces, while HC and exhaust gas temperature increases
Effect of engine speed	
Mansour et al. [12]	A light power and torque losses with regard to the diesel one, except around the speed of 2400 rpm. Slightly higher equivalence ratios for a given speed condition. Maximum combustion pressure is slightly higher for all engine speeds than the diesel fuelling level
Selim [14]	Pressure rise rate decreases with increase in engine speed and is higher than that for diesel case
Selim [4]	η_{th} improves with increasing engine speed. The pressure rise rate decreases as the engine speed increases for the three dual-fuel cases
Effect of pilot fuel injection timing	
Nwafor [2]	Offers higher BSFC and lower power output. Lower BTE with both advanced and standard timing compared to diesel case. Standard dual timing shows longer delay periods at high loads than the advanced injection timing operation. Higher HC emissions. High exhaust temperatures with advanced timing
Selim [14]	Pressure rise rate is higher than that for the 100% diesel engine as the injection advance increases (25–40° BTDC). For late injection of pilot (20–25° BTDC), combustion noise is comparable for diesel and dual-fuel cases
Abd et al. [33]	An improvement in thermal efficiency is achieved by advancing the injection timing. Advancing the injection timing at medium and high loads led to early knocking. Increase in NO_x, reduction in CO and UBHC emissions with advance injection timing
Selim [4]	Advancing the pilot fuel injection timing reduces the torque output and the thermal efficiency but, it increases the maximum pressure and maximum pressure rise rate
Krishnan et al. [35]	Fuel conversion efficiency increases with injection timing of 15–45° BTDC and decreases afterwards. Peak heat release rates are decreased with retard injection timing. Higher NO_x emissions for timing 45° BTDC than those for very retarded (15° BTDC) or very advanced (60° BTDC) timings
Effect of mass of pilot fuel inducted	
Badr et al. [9]	Exhaust emissions of the CO and the unconverted CH_4 are unaffected by the pilot quantity within a limiting equivalence ratio
Abd et al. [36]	An improvement in thermal efficiency by increasing the amount of pilot fuel. Increasing the amount of pilot fuel at high loads leads to early knocking. Increase in NO_x, reduction in CO and UBHC by increasing the amount of pilot fuel

Table 8 (*Continued*)

Researcher(s)	Performance of dual-fuel diesel engine
Selim [14]	Increasing the pilot fuel diesel mass increases the engine torque. The maximum pressure and the maximum rate of pressure rise is minimum for a pilot fuel mass of 0.52 kg/h, and it is increased for a lower or higher amount of diesel pilot fuel
Nwafor [37]	Increasing the pilot fuel and reducing primary fuel reduces the knocking phenomena
Selim [4]	Increasing the quantity of pilot diesel fuel increases the torque output and thermal efficiency for all fuels. Increasing the pilot fuel mass results in higher maximum combustion pressure but reduced maximum pressure rise rate
Effect of engine compression ratio	
Selim [4]	Knock starts earlier when a high compression ratio is used in the dual-fuel engine, and this is more notable for LPG. Increasing the compression ratio generally increases the combustion noise
Effect of engine intake manifold conditions	
Kusaka et al. [39]	EGR with intake heating improves thermal efficiency. EGR controls the rate of pressure rise. Excessive EGR ratio (>50%) causes the deterioration of combustion characteristics. EGR with intake heating reduces THC and NO_x emissions
Effect of type of gaseous fuel	
Bari [40]	40% CO_2 in biogas do not deteriorate the engine performance much as compared to the engine with NG (96% methane). But, 30% CO_2 in biogas improves the engine performance as compared to the same running with NG
Henham and Makkar [3]	Overall efficiency falls with gas mixture substitution and adding CO_2 affects this more at higher speed. 60% gasoil substitution is possible without knock. Exhaust temperature is affected more by NG substitution than by CO_2 addition except at maximum NG substitution. CO is affected mainly by NG substitution and less by gas quality. More rapid pressure rise on combustion with dual-fuel operation
Papagiannakis and Hountalas [24]	At high load slope of 'bsfc' increment with NG mass ratio is lower compared to the one at part load. Cylinder pressure decreases with increasing in the amount of NG. Peak cylinder pressure decreases with increase in quantity of gaseous fuel at constant load. Combustion duration increases with increasing NG mass ratio. Sharp decrease in soot concentration, lower NO, and high HC and CO emissions with increase in NG mass ratio
Selim [4]	Increasing the mass of gaseous fuel increases the combustion noise and maximum pressure for all the three gaseous fuels

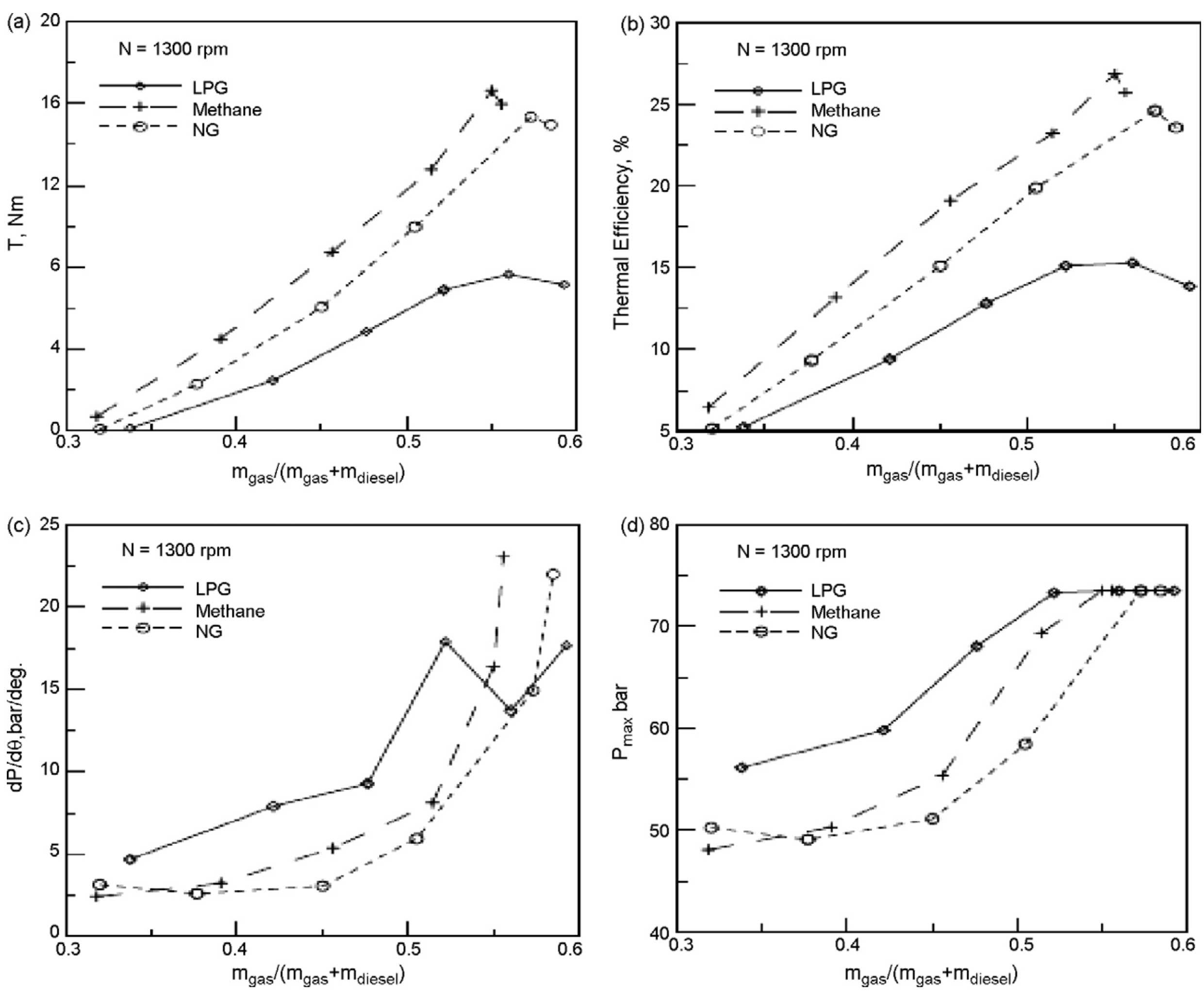

Fig. 90. Effects of mass of gaseous fuel used on performance and noise: N = 1300 rpm, IT = 35° BTDC, CR = 22, m_d = 0.37 kg/h [4].

emissions with increasing gaseous fuel mass ratio until a certain limit where they start to decrease. This is due to the increase of burnt gas temperature, which promotes the oxidation of UBHC. Under dual-fuel operation, the filling of the crevice volumes with unburned mixture of air and gaseous fuel during compression and combustion while the cylinder pressure continues to rise, is an important source dominating the formation of HC emissions. With a close look at Fig. 89, it is revealed that at part load, increasing the amount of gaseous fuel leads to a sharp increase of CO concentration. This is due to the slow combustion rate of gaseous fuel, which maintains the charge temperature at low levels resulting to a reduction of the oxidation process of CO. At high engine load, CO emissions increase with increasing NG mass ratio and beyond a certain value of gaseous fuel mass ratio they start to decrease, as a result of the high gas temperature and faster combustion rate. In general, CO emission values under dual-fuel operation are considerably higher compared to normal diesel operation.

Selim [4] has examined the effects of the amount of gaseous fuel, as a fraction of the total amount of fuel, in a dual-fuel diesel engine. During these experiments the constant parameters are; engine speed 1300 rpm, pilot fuel injection timing 35° BTDC, mass of pilot fuel 0.37 kg/h, compression ratio 22 and the amount of liquid diesel fuel.

Fig. 90a shows that for all the three gases the output torque increases with increasing the amount of gaseous fuel. It is noticed that the output torque and the thermal efficiency for the dual-fuel engine using pure methane is higher than that of the NG mixture which is higher than LPG. This is due to the higher LHV for methane (50 MJ/kg) compared to the natural gas (47.7 MJ/kg) mixture and LPG (46.1 MJ/kg). The dual-fuel engine, for all fuels used, however, suffers from low thermal efficiency at part load, and then it increases with increasing load by increasing the mass of gaseous fuel admitted (Fig. 90b). As the amount of gaseous fuel increases, the maximum combustion pressure and pressure rise rate increase for all three gaseous fuels (Fig. 90c). Increasing the load at constant speed resulted in an increase in the mass of gaseous fuel admitted to the engine, since the pilot mass injected is constant at all loads. This increase in the mass of gaseous fuel causes an increase in the ignition delay period of the pilot diesel. Then, the pilot fuel auto-ignites and starts burning the gaseous fuel at a higher rate of pressure rise. LPG produces the highest pressure rise rate as compared to methane and the NG mixture prior to knocking, Fig. 90c, because of its high tendency to self-ignite, and produce knocking combustion. The maximum pressure for the LPG case also appears to be the highest, followed by methane and then NG mixture (Fig. 90d). This is due to the early ignition of the LPG that produces higher maximum pressure BTDC, which tends to reduce the torque output produced for LPG (Fig. 90a).

The above literature studies, by different researcher(s), have revealed the effect of engine parameters and type of gaseous fuel on the performance of different dual-fuel diesel engines. The comparison of results concluded by these researcher(s) from their experimental programs is summarised in Table 8.

6. Conclusion

Researchers in various countries have carried out many experimental works using gaseous fuels as diesel engine fuel substitute in a dual-fuel mode of operation. An attempt has been made here to review the previous studies on dual-fuel concept. The overall observation from these experimental results is that the engine operating and design parameters, namely, load, speed, pilot fuel injection timing, pilot fuel mass, compression ratio, inlet manifold conditions, and type of gaseous fuel play an important role in the performance of dual-fuel diesel engines. Some of the salient points showing the effect of above listed parameters on the performance of dual-fuel engines are listed below.

6.1. Effect of engine load

- The dual-fuel engine performance decreases at part load conditions. There is a minor reduction in power output and higher BSFC for the engines.
- Lower peak cylinder pressure is for a dual-fuel engine compared to the normal diesel engine at a given load condition, which is encouraging since no danger exists for the engine structure. Pressure rise rate ($dP/d\theta$) increases with increase in load and is always higher than that of diesel fuel case.
- Combustion duration is longer compared to diesel operation at low load.
- Lower NO_x and drastic decrease in soot emissions with all gaseous fuels. But, at all load conditions, CO and HC emissions are considerably high compared to the diesel case.

6.2. Effect of engine speed

- Thermal efficiency improves with increasing engine speed. Slightly higher equivalence ratios for a given speed condition of dual-fuel engines.
- Maximum combustion pressure is slightly higher than the diesel fuelling level at constant engine speed.
- Pressure rise rate decreases with increase in engine speed and is higher than that for diesel case.

6.3. Effect of pilot fuel injection timing

- An improvement in thermal efficiency is achieved by advancing the injection timing.
- Maximum pressure and pressure rise rate is higher for the advanced injection timing compared with diesel case.
- Advancing the injection timing at medium and high loads led to early knocking.
- Increase in NO_x, and a reduction in CO and UBHC emissions with advance injection timing.

6.4. Effect of mass of pilot fuel inducted

- There is an improvement in thermal efficiency and torque output by increasing the amount of pilot fuel.
- Increasing the pilot fuel mass results in higher maximum combustion pressure but reduced maximum pressure rise rate.
- Early knocking with increase in the amount of pilot fuel at high loads.
- Increasing the pilot fuel and reducing primary fuel reduces the knocking phenomena.
- Higher NO_x and reductions in CO and UBHC by increasing the amount of pilot fuel.

6.5. Effect of engine compression ratio

- Knock starts earlier when a high compression ratio is used.
- Increasing the compression ratio generally increases the combustion noise.

6.6. Effect of engine intake manifold conditions

- EGR with intake heating improves thermal efficiency.
- Excessive EGR ratio (>50%) causes the deterioration of combustion characteristics.

- EGR with intake heating reduces THC and NO_x emissions.

6.7. Effect of type of gaseous fuel

- The engine performance is not deteriorated much with 40% CO_2 in biogas as compared to the engine with NG (96% methane). But, 30% CO_2 in biogas improves the engine performance as compared to the same running with NG.
- 60% gasoil substitution is possible by gas mixture without knock.
- Overall efficiency falls with gas mixture substitution and adding CO_2 affects this more at higher speed.
- Sharp decrease in soot concentration, lower NO, and high HC and CO emissions with increase in NG mass ratio.
- Increasing the mass of gaseous fuel increases the combustion noise and maximum pressure for methane, CNG and LPG.

It seems that dual-fuel combustion using gaseous fuels is a promising technique for controlling both NO and soot emissions even on existing diesel engines with slight modification to the engine structure. The penalty in 'bsfc' experienced is partially compensated by the lower price of gaseous fuels. The observed disadvantages, at low engine load condition, concerning 'bsfc', HC and CO can be reduced by applying modifications in engine tuning, i.e. injection timing of the pilot fuel. Again, in diesel dual-fuel engines the ignition characteristics of the gaseous fuels are still to be understood and needs more research on it. Thus, in overall, the engine operating and design parameters, and selection of type of gaseous fuel has to be chosen accordingly for an existing diesel engine to run on dual-fuel concept. This can minimize the engine performance, combustion and emission characteristics divergences between the existing diesel engine and a dual-fuel diesel engine.

Acknowledgements

The authors wish to thank and acknowledge the following organizations for permission to reproduce the figures from their publications in this paper: The American Society of Mechanical Engineers (USA), Elsevier (UK), Society of Automotive Engineers (Japan), The Institution of Engineers (India), and Indian Academy of Sciences (India).

References

[1] Nwafor OMI. Effect of choice of pilot fuel on the performance of natural gas in diesel engines. Renewable Energy 2000;21(3/4):495–504.
[2] Nwafor OMI. Effect of advanced injection timing on the performance of natural gas in diesel engine. Sadhana 2000;25(1):11–20.
[3] Henham A, Makkar MK. Combustion of simulated biogas in a dual-fuel diesel engine. Energy Conversion and Management 1998;39(16–18):2001–9.
[4] Selim MYE. Sensitivity of dual fuel engine combustion and knocking limits to gaseous fuel composition. Energy Conversion and Management 2004;45(3): 411–25.
[5] White TR, Milton BE, Behnia M. Direct injection of natural gas/liquid diesel fuel sprays. In: Proceedings of the 15th Australasian fluid mechanics conference; 2004.
[6] Liu Z. An examination of the combustion characteristics of compression ignition engines fuelled with gaseous fuels. Ph.D. thesis. Canada: University of Calgary, Department of Mechanical Engineering; 1995.
[7] Karim GA. The dual fuel engine. In: Evans RL, editor. Automotive engine alternatives. New York: Plenum Press; 1987.
[8] Thyagarajan V, Babu MKG. A combustion model for a dual fuel direct injection diesel engine, diagnostics and modelling of combustion in reciprocating engine. In: Proceedings of the Comodia symposium; 1985.p. 607.
[9] Badr O, Karim GA, Liu B. An examination of the flame spread limits in a dual fuel engine. Applied Thermal Engineering 1999;19(10):1071–80.
[10] Turner SH, Weaver CS. Dual fuel natural gas/diesel engines: technology, performance and emissions. Gas Research Institute Technical Report no. 94/ 0094; 1994.
[11] Heywood JB. Internal combustion engine fundamentals. New York: McGraw-Hill Book Co.; 1988.
[12] Mansour C, Bounif A, Aris A, Gaillard F. Gas–diesel (dual-fuel) modelling in diesel engine environment. International Journal of Thermal Science 2001; 40(4):409–24.
[13] Nwafor OMI. Combustion characteristics of dual-fuel diesel engine using pilot injection ignition. Institution of Engineers (India) Journal April 2003; 84:22–5.
[14] Selim MYE. Pressure–time characteristics in diesel engine fuelled with natural gas. Renewable Energy 2001;22(4):473–89.
[15] Nielsen OB, Qvale B, Sorenson S. Ignition delay in the dual fuel engine. SAE paper 870589; 1987.
[16] Uma R, Kandpal TC, Kishore VVN. Emission characteristics of an electricity generation system in diesel alone and dual fuel modes. Biomass & Bioenergy 2004;27(2):195–203.
[17] Parikh PP, Bhave AG, Kapse DV, Shashikantha. Study of thermal and emission performance of small gasifier-dual-fuel engine systems. Biomass 1989;19:75–97.
[18] Sridhar G, Paul PJ, Mukunda HS. Biomass derived producer gas as a reciprocating engine fuel—an experimental analysis. Biomass & Bioenergy 2001;21(1): 61–72.
[19] Papagiannakis RG, Hountalas DT. Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas. Energy Conversion and Management 2004;45(18/ 19):2971–87.
[20] Hountalas DT. The effect of operating parameters on the net and gross heat release rates of a direct injection diesel engine. In: Proceedings of the second biennial ASME-ESDA international conference on design of energy systems; 1994. 63(3).
[21] Benson RS, Whitehouse ND. Internal combustion engines. Oxford: Pergamon Press; 1973.
[22] Lavoie GA, Heywood JB, Keck JC. Experimental and theoretical study of nitric oxide formation in internal combustion engines. Combustion Science Technology 1970;1:313–26.
[23] Kouremenos DA, Rakopoulos CD, Hountalas DT. Experimental investigation of the performance and exhaust emissions of a swirl chamber diesel engine using JP-8 aviation fuel. In: International journal of energy research, vol. 21. John Wiley & Sons; 1997.
[24] Papagiannakis RG, Hountalas DT. Experimental investigation concerning the effect of natural gas percentage on performance and emissions of a DI dual fuel diesel engine. Applied Thermal Engineering 2003;23(3): 353–65.
[25] Karim GA, Khan MO. Examination of effective rates of combustion heat release in a dual-fuel engine. Japan Society of Mechanical Engineers 1968;10.
[26] Agarwal A, Assanis DN. Multidimensional modelling of natural gas ignition under compression ignition conditions using detailed chemistry. SAE Paper 980136; 1998.
[27] Karim GA. A review of combustion processes in the dual fuel engine-the gas diesel engine. Progress in Energy and Combustion Science 1980;6(3):277–85.
[28] Liu Z, Karim GA. Simulation of combustion processes in gas-fuelled diesel engines. Proceedings of Institution of Mechanical Engineers 1977;211(A): 159–69.
[29] Hountalas DT, Papagiannakis RG. Development of a simulation model for direct injection dual fuel diesel—natural gas engines. SAE Trans. 2000-01-0286; 2000.
[30] Singh RN, Singh SP, Pathak BS. Investigations on operation of CI engine using producer gas and rice bran oil in mixed fuel mode. Renewable Energy 2007;32(9):1565–80.
[31] Imoto K, Sugiyama S, Fukuzawa Y. Reduction of combustion-induced noise in IDI diesel engine (1st report, target of cycle and low-noise combustion). Transactions of Japan Society of Mechanical Engineers Part B 1997;63(605): 329–35.
[32] Selim MYE. Effect of engine parameters and gaseous fuel type on the cyclic variability of dual fuel engines. Fuel 2005;84(7/8):961–71.
[33] Abd Alla GH, Soliman HA, Badr OA, Abd Rabbo MF. Effect of injection timing on the performance of a dual fuel engine. Energy Conversion and Management 2002;43(2):269–77.
[34] Abdalla AY, Radwan MS, Ahmed SH. Smoke level and operational roughness of a pre-combustion chamber diesel engine running on gasoil/methanol blends. SAE Paper 912358; 1991.
[35] Krishnan SR, Srinivasan KK, Singh S, Bell SR, Midkiff KC, Gong W, et al. Strategies for reduced NO_x emissions in pilot-ignited natural gas engines. Journal of Engineering for Gas Turbines and Power 2004;126:665–71.
[36] Abd Alla GH, Soliman HA, Badr OA, Abd Rabbo MF. Effect of pilot fuel quantity on the performance of a dual fuel engine. Energy Conversion and Maagement 2000;41(6):559–72.
[37] Nwafor OMI. Knock characteristics of dual-fuel combustion in diesel engines using natural gas as primary fuel. Sadhana 2002;27(3):375–82.
[38] Nwafor OMI, Rice G. Combustion characteristics and performance of natural gas in high speed indirect injection diesel engine. Renewable Energy 1994; 5(5–8):841–8.
[39] Kusaka J, Okamoto T, Daisho Y, Kihara R, Saito T. Combustion and exhaust gas emission characteristics of a diesel engine dual-fueled with natural gas. Society of Automotive Engineers of Japan Review 2000;21(4):489–96.
[40] Bari S. Effect of carbon dioxide on the performance of biogas/diesel duel-fuel engine. Renewable Energy 1996;9(1–4):1007–10.

[41] Chigier N. Energy, combustion, and environment. USA: McGraw Hill Company; 1981.
[42] Zhanghou J. Utilization and distribution of biogas (memeograph). China: Tongi University; 1990.
[43] Hountalas DT. Available strategies for improving the efficiency of DI diesel engines—a theoretical investigation. SAE Paper 2000-01-1176; 2000.
[44] Rakopoulos CD, Hountalas DT, Taklis GN, Tzanos EI. Analysis of combustion and pollutants formation in a direct injection diesel engine using a multi-zone model. International Journal of Energy Research 1995;19(1):63–88.
[45] Kouremenos DA, Hountalas DT, Kouremenos AD. Experimental investigation of the effect of fuel composition on the formation of pollutants in direct injection diesel engines. SAE paper 1999-01-0527; 1999.